Absorption chillers: their feasibility in district heating networks and comparison to alternative technologies

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Abstract

In recent years, a rising cold demand can be observed, which is e.g. due to increased requirements on climate control. The majority of this cold is currently provided by compression chillers. At the same time, district heating companies exhibit a lack of sales during summertime, which could e.g. be diminished by the operation of thermal driven cooling systems.
This work assesses absorption chiller technology in combination with boundary conditions implied by district heating systems. A theoretical comparison with compression chillers is provided and thereby interesting technology is identified. An analysis of the respective absorption chiller market in Germany is performed and a real case cold generation system is evaluated. Furthermore, the economic competitiveness to compression chillers is considered and environmental footprints are compared.
The LiBr-water single-effect absorption chiller is identified as the only reasonable machine, to be driven by district heat. Factors of highest influence are load factors, prices for heat, efficiency and size of the chillers. Possible application fields can be found e.g. in large office buildings, including server rooms. Absorption chillers are less appropriate for the use as stand alone systems and fluctuating load profiles, but conceivable as complementary to compression chillers, providing a constant base load. Buffer storages are essential for high load factors, continuous operating conditions and consequently efficiency. The calculated acceptable price for heat is significantly lower than typical market prices for district heat. Nevertheless, the provision of the product "cold" under agreed-upon conditions is conceivable as business model for district heating companies.
Keywords: *air-conditioning, absorption chiller, district heating, real case, market overview, economic analysis*

Resumo

Nos últimos anos tem-se verificado um aumento do consumo de energia para arrefecimento devido, por exemplo, à melhoria das condições de climatização dos edifícios. Atualmente, a quase totalidade da produção de frio para estes fins é assegurada por sistemas elétricos de compressão de vapor. Por outro lado, as companhias que exploram as redes de aquecimento urbano têm uma quebra de negocio durante o verão que poderia ser mitigada pelo uso de sistemas de absorção para produzir frio durante esta estação do ano.

No presente trabalho estuda-se a exequibilidade da utilização alternativa de sistemas de absorção alimentados por redes de calor urbano e identifica-se qual a melhor tecnologia para este fim. É analisado o mercado alemão de equipamentos e as especificações de um fabricante são comparadas com os resultados obtidos com um sistema real. Analisa-se ainda competitividade económica destes sistemas e o seu impacto ambiental.

Os sistemas de efeito simples com LiBr/H_2O foram identificados como a única tecnologia competitiva para a finalidade pretendida. Os factores que mais influenciam esta competitividade são os fatores de carga, o preço do calor, a eficiência dos sistemas e a dimensão das máquinas. Devido ao seu pior desempenho dinâmico estes sistemas são mais adequados como complemento de sistemas de compressão de vapor, fornecendo uma potência de base constante.

Os preços do calor competitivos são significativamente inferiores aos preços típicos do calor dos sistemas de aquecimento urbano. No entanto, a produção e fornecimento de frio em condições pré-contratadas é um modelo de negócio exequível para estas empresas.

Palavras chave: *ar-condicionado, refrigeração por absorção, redes de aquecimento urbano, caso real, mercado atual, análise económica*

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U	[kJ]	Internal energy
Q	[kJ]	Heat
W	[kJ]	Work
Q_0	[kJ]	Cooling power
Q_C	[kJ]	Rejection heat
Q_{gen}	[kJ]	Supplied heat in the generator
Q_{abs}	[kJ]	Rejected heat in the absorber
$P_e l$	[kJ]	Electrical power
w	[kJ/kg]	Specific work
w_t	[kJ/kg]	Specific technical work
q	[kJ/kg]	Specific heat
s	[kJ/kg]	Specific entropy
h	[kJ/kg]	Specific enthalpy
η_{compr}	[-]	Isentropic efficiency compressor
$\dot{m}$	[kg/s]	Mass flow
x	[-]	Mass fraction absorbency
p	[bar]	Pressure
c	[kJ/kgK]	Specific heat capacity water
COP	[-]	Coefficient of performance
$COP_{CarnotHP}$	[-]	Coefficient of performance heat pump
$COP_{CarnotCM}$	[-]	Carnot efficiency compression chiller
$COP_{CarnotAM}$	[-]	Carnot efficiency absorption chiller
COP_{rated}	[-]	Coefficient of performance for rated conditions
T	[$^\circ$C]	Temperature
T_C	[$^\circ$C]	Condensing temperature
T_0	[$^\circ$C]	Vaporizing temperature
T_A	[$^\circ$C]	Absorption temperature
T_{gen}	[$^\circ$C]	Temperature in the generator
$T_{recool,in}$	[$^\circ$C	Temperature of the cooling water, entering the chiller
$T_{recool,out}$	[$^\circ$C]	Temperature of the cooling water, leaving the chiller
$T_{cold,in}$	[$^\circ$C]	Temperature of the cold water, entering the evaporator
$T_{cold,out}$	[$^\circ$C]	Temperature of the cold water, leaving the evaporator
$T_{heat,in}$	[$^\circ$C]	Temperature of the hot water, entering the generator
$T_{heat,out}$	[$^\circ$C]	Temperature of the hot water, leaving the generator
V	[m^3]	Volume
V_{BS}	[m^3]	Volume of the buffer storage
f_C	[-]	Price index cooling system components

f_p	[-]	Primary energy factor
β	[-]	CO_2 equivalency factor
C_{CM}	[€/kW]	Investment costs compression machine
C_{AM}	[€/kW]	Investment costs absorption machine
C_{CT}	[€/kW]	Investment costs cooling tower
C_{BS}	[€/kW]	Investment costs buffer storage
A_N	[€/kW]	Annuity
$A_{N,C}$	[€/kW]	Annuity for capital-related costs
$A_{N,D}$	[€/kW]	Annuity for demand-related costs
$A_{N,Op}$	[€/kW]	Annuity for operation-related costs
$A_{N,O}$	[€/kW]	Annuity for other costs
b	[€/kW]	Price dynamic cash value factor
a	[€/kW]	Annuity factor
R_W	[€]	Residual value
p_{Cold}	[€/kW]	Specific costs for cold generation
$P_{Cold,a}$	[kWh/a]	Annual cold production

1 Introduction

In recent years the energy sector is undergoing a radical transition, which affects its general power generation and distribution structures. Branches and corresponding infrastructures, that have been developed successfully over many years have suddenly faced big challenges in order to comply with new requirements, implied by these new structures. The awareness of the finiteness of conventional energy carriers as well as the global warming and its associated negative consequences on human habitat has lead to a change in mindsets. States all over the world define common climate goals like in the Kyoto Protocol and commit themselves to limit emissions and improve resource-effectiveness. As a result, previous corner pillars of energy production are questioned and options are explored by which the set goals can be achieved. As a consequence of the nuclear disaster in Fukushima in 2011, Germany agreed on a progressive and complete phase-out of nuclear energy, which leads to the disappearance of large supply capacities. The resulting energy gap needs to be closed by obeying the regulations set by the authorities but at the same time being cost effective and even more important, maintaining the security of supply.

In the recent past, this gap was mainly closed by the addition of new renewable capacity. Renewables, such as solar power and wind power, however depend on external conditions such as the availability of wind speed and solar radiation. They are fluctuating in time and thus are not controllable as it is the case for conventional and nuclear systems. In view of the above, an energy system solely consisting of renewables is not very likely to fulfill the security of supply in the near future. For this reason, existing conventional power plants like highly modern coal-fired and gas-fired power plants still play an important role and will also be essential for the success of the energy mix in the near future. These conventional power plants, which currently produce more than 50% of the power required in Germany, are always available and therefore ensure service security. A severe disadvantage of such plants is however the emission of greenhouse gases such as CO_2 which goes along with the production of energy. A smart way to reduce the ratio between emissions and provided energy is the upgrading of existing power plants to so called Combined Heat and Power plants (CHP). Thereby the energy efficiency can be increased by enabling simultaneous generation of usable heat and power. For the reason, that existing plants can be used, the retrofitting is a cost-effective variant in order to reach climate goals.

In the winter months, currently both heat and power can be provided economically through CHP generation and about 90% of district heat is provided by cogeneration in several German cities such as Stuttgart. In summer however, there is a discrepancy between the actual demand and the heat that could be produced if the CHP plants would be operated in an optimal working point. This lack of demand for heat in summer represent the ultimate obstacle for further extension of cogeneration respectively an even better utilization of existing units.

In the recent years, a remarkable increase in cold demand can be observed in Germany and several other industrial countries, which can be traced back e.g. to higher demands on climate control in office buildings and an increasing amount of data centers. This market request is currently mainly satisfied by compression chillers, which is the currently dominating refrigeration technology in market.

However, to be more efficient concerning emissions, thermal driven cooling systems such as absorption chillers, in combination with an increased number of CHP power plants could possibly be a better option. In contrast to compression chillers, which use electric energy in order to compress a cooling agent, absorption chillers use heat for this process, generating thereby the desired cold. Hence in summer, the demand for cold could be satisfied and at the same time the demand for electricity could be kept low by avoiding the installation of additional electrically operated compression chillers.

In this work, the technological, economic and environmental maturity of thermal driven refrigeration technologies, in particular of absorption chillers, is assessed as an competitive alternative to compression chillers.

In chapter 2, the underlying fundamentals, working principles and properties of the different technologies are introduced in order to compare the compression chiller as dominant system on the market to the thermal driven systems. In chapter 3 potential application fields for absorption chillers are identified, considering the boundary conditions that are implied by the use of district heat and to some extent also decentralized CHP plants. The German absorption chiller market is analysed and an overview of suitable machines is given in chapter 4. A manufacturer inquiry is conducted, as working parameters and efficiencies, freely available or given by the different manufacturers, do not necessarily underly uniform assumptions. Furthermore these parameters are at the same time highly dependent on these specific boundary conditions. For verification of the theoretical data and indications, a real case example of an air-conditioning concept with a state of the art absorption chiller is evaluated in chapter 5. In addition the economic efficiency of absorption chillers and the competitiveness to compression machines is evaluated by comparing the costs for cold generation in various scenarios as well as identifying the sensitivities of the different technologies (chapter 6). In a last step environmental indicators are calculated and compared between the technologies (chapter 7).

2 Cooling technologies

The purpose of a cooling machine is to ensure that the temperature of a certain system (e.g. a room) is lowered in contrast to the temperature of its environment. As cold cannot be produced according to the second law of thermodynamics, heat has to be removed from the system and dispatched to another system. In summary, a refrigerator absorbs heat from the system to be cooled, transports it and returns it to a different system [1]. There exist several different methods and technologies that are capable of providing a cooling effect, such as cold vapor machines, cold gas machines as well as electric and magnetic processes.

Since possibilities to increase the utilization of heat shall be found, thermal driven systems are of particular importance in this work. Thereby it is distinguished between open systems, using the process of adiabatic air humidification in order to provide the cooling effect and closed systems. In this work only closed cold vapor machines are taken into consideration. Thereby the main focus is on the technology that has the highest market maturity and that is suitable for the use of district heat, absorption chillers. Like previously explained, the compression chiller is examined as well in order to be able to compare the systems.

This chapter introduces the different cold vapor machines and compares them within each other.

2.1 Compression chiller

The compression chiller is nowadays the most common chiller in market.

A single stage compressor machine in its basic structure is shown in figure 2.1. The four most important components are the two heat exchangers, one absorbing heat at low temperatures, providing the cooling power Q_0 (evaporator) and one which is releasing the heat Q_C at high temperatures to the environment (condenser). In the evaporator, the liquid cooling agent is vaporized by the absorption of heat. The vapor is then sucked by the compressor and compressed by applying mechanical work. At this higher pressure, the refrigerant is condensed in the condenser and afterwards expanded in an expansion valve to its initial state.

2.1.1 Thermodynamic cycle

The implementation of the described refrigeration process is realized in a thermodynamic cycle. A thermodynamic cycle is a process chain in which a working fluid passes several changes of state by the supply or removal of heat and work. At the end of this process chain the fluid returns to its initial state. In general it is distinguished between clockwise (e.g. heat engine) and counter-clockwise operated cycles. Counter-clockwise operated cycles are used in heating and cooling applications.

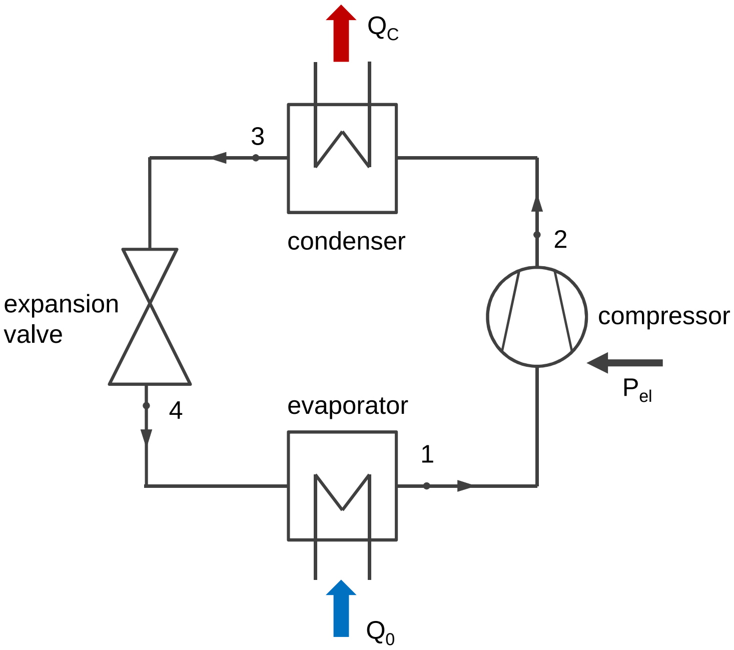

Figure 2.1: Basic Working scheme of a single stage compression chiller

The typical counter-clockwise operated cycle consists of the following four basic operations:

- Absorption of heat at low temperature
- Increase of temperature and pressure
- Release of heat at higher temperature
- Expansion to the initial state

Whether the machine can be regarded as a heat pump or a chiller depends on the point of view respectively the useful space which is to be climatized. In a heat pump application the used system is connected to the heat exchanger that releases the heat at the high temperature level (condenser). In a cooling application the system which needs to be cooled provides the heat in the low temperature heat exchanger (evaporator).

2.1.2 The ideal carnot cycle

The carnot process is the basic thermodynamic cycle of both heating engines (clockwise), heat pumps and cooling machines (counter-clockwise). The process solely consists of reversible substeps and is therefore often used as a reference cycle in order to evaluate performances of real cycles. The process consists of the same four basic operations as described above, however they are conducted in a certain optimized way [2]:

- $4 \rightarrow 1$: Isothermal absorption of heat

$$q_{41} = q_{in} = T_0(s_1 - s_4); w_{41} = (h_1 - h_4) - T_0(s_1 - s_4) = 0 \tag{2.1}$$

- $1 \rightarrow 2$: Isentropic compression

$$q_{12} = 0; w_{12} = h_2 - h_1; \Delta s_{12} = 0 \tag{2.2}$$

- $2 \rightarrow 3$: Isothermal release of heat

$$q_{23} = T_C(s_3 - s_2); w_{23} = (h_3 - h_2) - T_C(s_3 - s_2) = 0 \tag{2.3}$$

- $3 \rightarrow 4$: Isentropic expansion

$$q_{34} = 0; w_{34} = h_4 - h_3; \Delta s_{34} = 0 \tag{2.4}$$

The energetic performance of a compression chiller is defined by the coefficient of performance (COP), which is calculated as ratio between benefit and expenditure.

$$COP = \frac{benefit}{expenditure} \tag{2.5}$$

With the help of the T-s diagram of a carnot cycle (figure 2.2), equation 2.5 can be well illustrated. The benefit in case of a cooling machine is the heat that is absorbed from a system at low temperature level (q_0). According to the second law of thermodynamics this heat is

$$dq_0 = T_0 \cdot ds \tag{2.6}$$

Graphically this heat can be depicted by the green area below the line 4-1 and is

$$q_0 = (s_1 - s_4) \cdot T_0 \tag{2.7}$$

The heat that is released to the environment can respectively expressed by both the green and the yellow area and is

$$q_C = (s_3 - s_2) \cdot T_C \tag{2.8}$$

In order to get the refrigerant to the required high pressure and temperature at which the heat is released to the other system, technical work (w_t) is required. This specific technical work can be calculated to

$$w_t = (s_2 - s_3) \cdot T_C - (s_1 - s_4) \cdot T_0 \tag{2.9}$$

and is therefore represented in the graph by the yellow area. Consequently the carnot performance of a compression chiller can be calculated by

$$COP_{CarnotCM} = \frac{T_0}{T_C - T_0} \tag{2.10}$$

For heat pump applications the ratio is usually called coefficient of performance (COP) and is respectively calculated by

$$COP_{CarnotHP} = \frac{T_C}{T_C - T_0} \tag{2.11}$$

It can be seen, that the carnot performance is solely dependent on the choice of the two temperature levels. Furthermore in order to increase process efficiency, heat should be absorbed (it should be cooled) at the highest possible temperature and released to the environment at the lowest possible temperature. It can be also seen, that the $COP_{CarnotCM}$ of a cooling machine is always $COP_{CarnotHP} - 1$.

2.1.3 Theoretical and real vapor compression cycle

The theoretical cooling cycle of a single stage compression machine is shown in figure 2.3 both in a T-s diagram and a log p-h diagram. As it can be seen, this cycle differs from the carnot cycle. This is

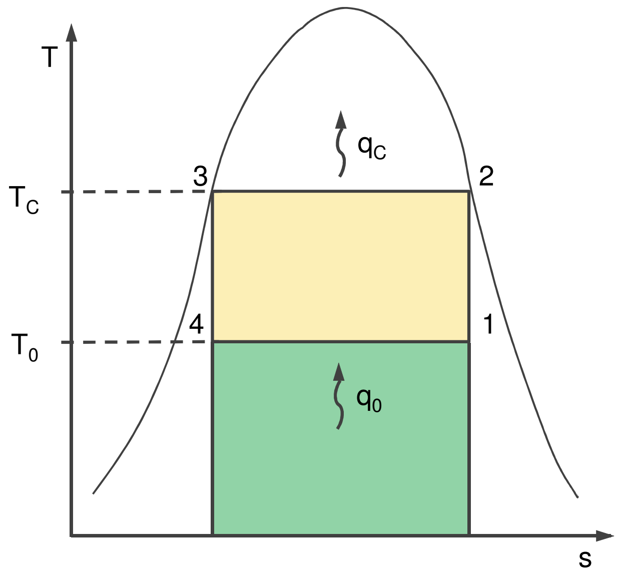

Figure 2.2: Carnot process in a T-s diagram

because it is not possible to operate the carnot cycle in reality, due to technical restrictions of the components and irreversibilities that cannot be overcome. In the log p-h diagram the net power q_0 and the required work w_t can be seen as lines and this diagram is therefore often used to visualize such cycles.

The cycle takes place between two pressure levels, the lower evaporator pressure p_0 and the higher condenser pressure p_c. In reality, those pressures are usually selected in a way that the corresponding boiling temperatures for evaporator pressure $T_S(p_0)$ is slightly lower than the temperature in the cooling chamber and the boiling temperature for condenser pressure $T_S(p_C)$ is higher than ambient temperature. This is because in order to make heat exchange possible, sufficient temperature differences between cooling agent and cooling chamber respectively cooling agent and ambient air need to be guaranteed (typically about 5K[1]).

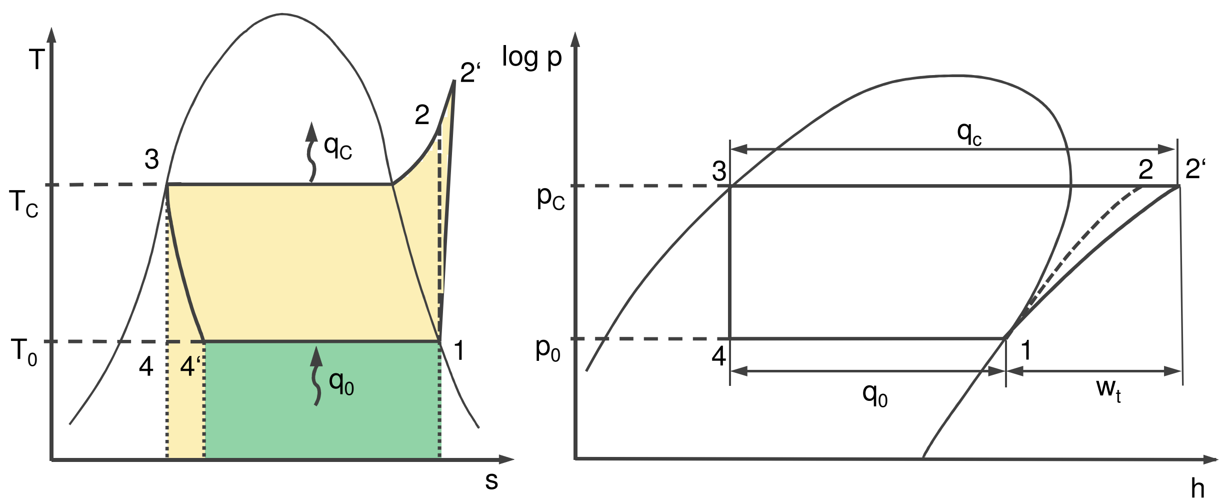

Figure 2.3: Theoretical vapor compression cycle incl. compressor losses in a T-s diagram and log p-h diagram (own graphic adapted from[2])

The compressor compresses the refrigerant vapor after leaving the evaporator (1). Depending on con-

[1] For a comprehensive introduction to the basics of heat transfer e.g. [3] is recommended.

struction and application it can be e.g. a reprocating piston, a rotary piston, like a scroll or a screw compressor as well as for higher cold performances a turbo compressor [4]. In real technical applications, the vapor at point (1) is slightly superheated in order to avoid harmful and sudden explosions in the compressor (not depicted in figure 2.3). As real processes are always irreversible, the process does not happen isentropically and therefore the real state point (2') has a higher enthalpy and entropy than the theoretical point (2). In the condenser, the superheated vapor is cooled down until it reaches the saturated vapor line at condensating temperature T_C (3). In real applications, the condensate is for safety reasons often cooled down for another 5K-10K (not depicted in figure 2.3) in order to guarantee vapor free liquid at the entry of the expansion valve[5]. Then the refrigerant is expanded isenthalpic to evaporator pressure and state (4') is reached.

The specific net cooling performance q_0 is represented in the log p-h diagram by the length of the line between (4) and (1) and the required specific work w_t by the distance on the abscissa between point (1) and (2'). The overall net power can consequently be calculated to

$$\dot{Q}_0 = \dot{m} \cdot (h_1 - h_4') = \dot{m} \cdot (h_1 - h_3) \tag{2.12}$$

with the cooling agent mass flow rate $\dot{m}$. The required driving power P_{el} is

$$P_{el} = \dot{m} \cdot w_t = \dot{m} \cdot (h_2' - h_1) = \frac{\dot{m}}{\eta_{compr}} \cdot (h_2 - h_1) \tag{2.13}$$

with η_{compr} being the isentropic efficiency of the compressor. The coefficient of performance is then calculated to

$$COP = \frac{\dot{Q}_0}{P_{el}} = \eta_{compr} \cdot \frac{h_1 - h_3}{h_2 - h_1} \tag{2.14}$$

So far just the losses in the compressor have been considered, nevertheless in reality all of the used components have losses. The compressor usually is the component with the highest losses, followed by the condenser, the expansion valve and the evaporator. Furthermore the pipes which connect the different components cause losses.[6]

2.1.4 Design possibilities of compression cycles

There exist numerous different forms for the realization of compression cooling cycles. Which cycle is the most adequate one is usually dependent on various different factors such as choice of the refrigerant, ambient temperature, required cooling temperature, etc.. In the following some of those possibilities are presented.

Single stage compression with (internal) superheating and supercooling

From figure 2.3 it can be derived, that the net cooling power could be increased by supercooling the water after condensation as well as by superheating the vapor in the evaporator. In technical applications both options can be found. The superheating is done in order to avoid sudden deflagrations in the compressor and is usually in the order of magnitude of 5K-10K. The increased compressor intake temperature nevertheless leads to disproportional high compression end-temperatures. Furthermore higher temperatures increase the volume at compressor intake state (1), what leads to a higher required work in the compressor. To what extend a superheating of the refrigerant at evaporator outlet is convenient, must be decided after an optimization process from case to case. Supercooling of the refrigerant after condensation however always leads to an increase of net cooling power without increasing the compression power at the same time and therefore increases the COP. However, depending on laid-out conditions, often very low cooling water temperatures are required in order to make supercooling possible. This can lead to inefficient operation of the heat sink system. [2]

An option to integrate both supercooling and superheating is an internal heat exchanger between outlet of the condenser and outlet of the evaporator as shown in figure 2.4.

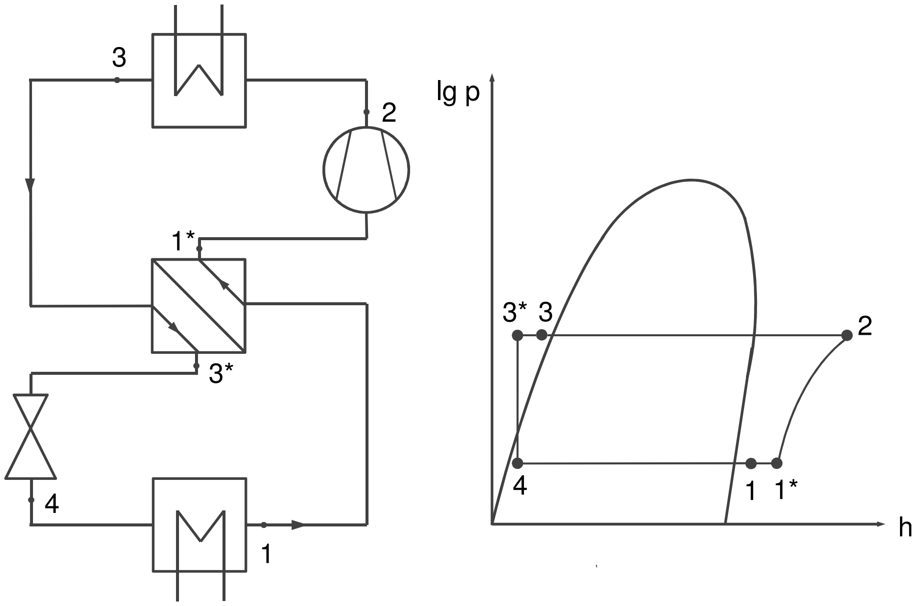

Figure 2.4: Single stage compression cycle with internal supercooling and superheating (own illustration according to [7])

Multiple stage compression chillers

The ratio between condenser pressure p_c and evaporator pressure p_0 increases with increasing condensing temperature and/or decreasing vaporization temperature (see figure 2.3). However, the pressure ratio is limited by technical and economical constraints of currently available compressors (see table 2.1. Technical limits are e.g. related to max. permissible temperatures of lubricants, compressor efficiencies etc..
A two stage compression should be considered if one or more of the following conditions are valid [7]:

- The maximum permissible compression temperature is exceeded in single stage operation
- The added piston displacement of both stages is bigger than for single stage operation for a certain performance
- The specific compression work is lower as for single stage operation

In order to decrease the compression temperature and compression work it is inevitable to cool the refrigerant in between the two compression stages. This cooling can be realized in different ways. In the following some common variants of two stage compression cycles are presented.

Two stage compression chiller with external intercooling

An external heat exchanger is integrated between the two compressors. Usually it is an ambient air or water heat exchanger whose minimal reachable temperature is limited by ambient air respectively cooling water temperature plus the for heat flow necessary temperature spreading. The mass flow is the same for both compressors.

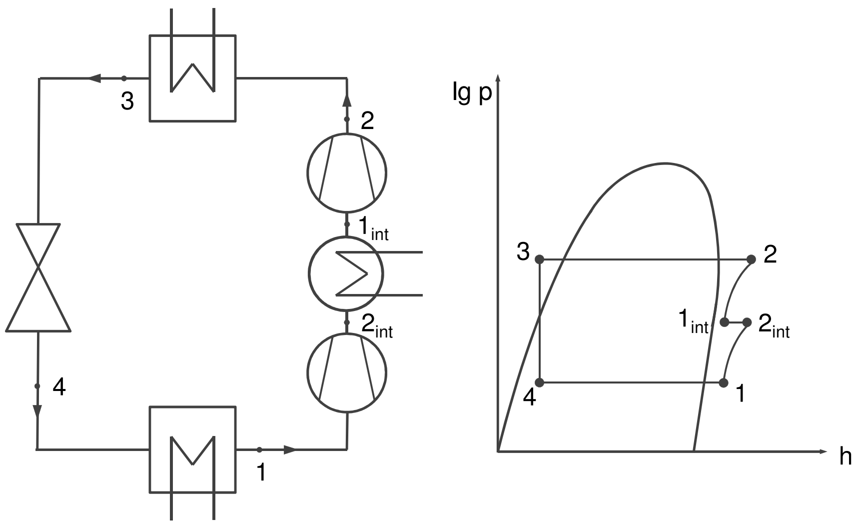

Figure 2.5: Working scheme and log p-h diagram of a two stage compression chiller with external intercooling (own illustration according to [7])

Two stage compression with intercooling via intermediate refrigerant injection

The injection of the cooling agent is realized by a thermostatic expansion valve. The setup has relatively low investment costs but does not increase the specific net power $h_1 - h_4$. In order to minimize the risk of sudden flash fires, due to insufficient vaporization of the cooling agent, an internal heat exchanger is often included before the mass flow enters the high pressure compressor.

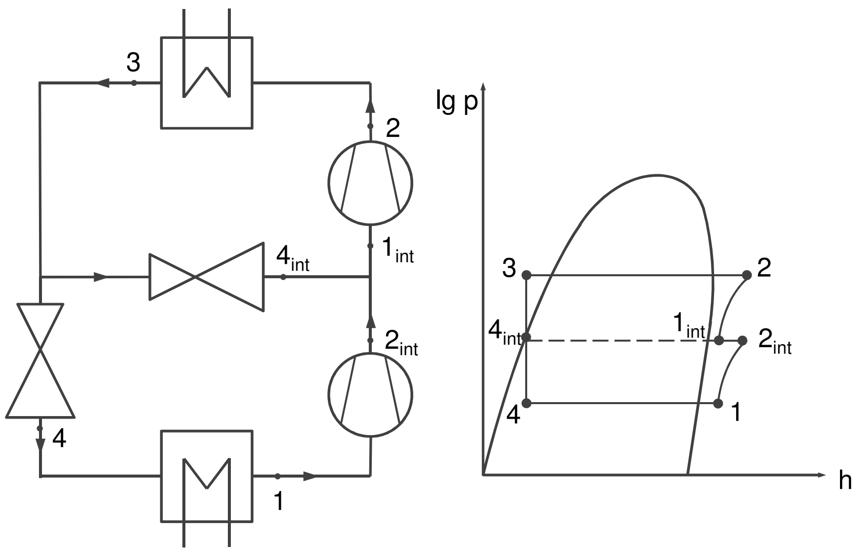

Figure 2.6: Working scheme and log p-h diagram of a two stage compression chiller with intercooling via intermediate refrigerant injection (own illustration according to [7])

Two stage compression with intercooling via intermediate pressure vessel

In this setup an intermediate pressure vessel is installed between the two compressor stages and the two expansion valves. It is filled with liquid refrigerant and is kept constant at a certain level by a nivel-controlled expansion valve. The liquid cooling agent is fed from there through the low pressure expansion valve to the compressor. After evaporation it is sucked by the low pressure compressor and is then pressed into the intermediate pressure vessel, causing a partial vaporization of the liquid cooling agent. Thereby the cooling to the intermediate pressure associated boiling temperature is reached.

This intercooling set-up is often used in rather large compression chillers, mainly driven by ammonia. Thereby a big increase in specific net power can be reached which is often considerably bigger than for systems in which the intercooling is done via external heat exchanger.

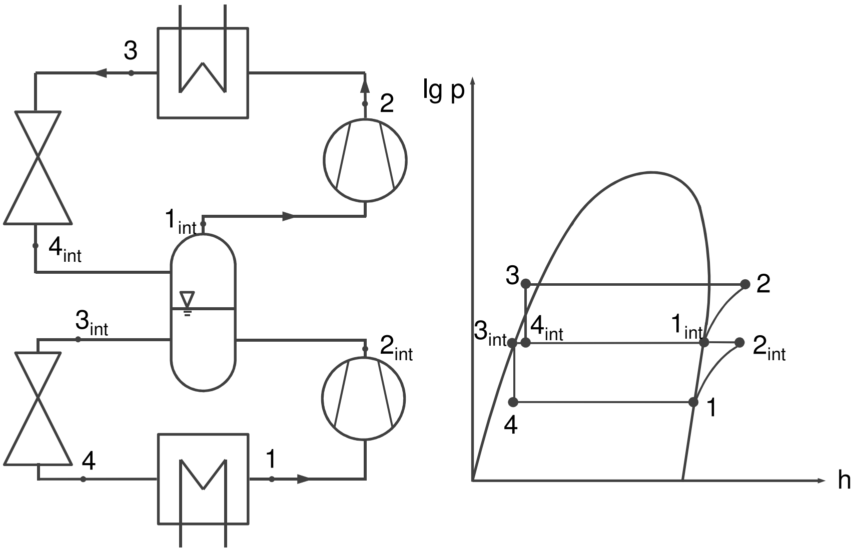

Figure 2.7: Working scheme and log p-h diagram of a two stage compression chiller with intercooling via intermediate pressure vessel (own illustration according to [7])

Cascade connection

In cascade connection, two (or more) different refrigerants are used in individual cycles that are connected via heat exchangers. The advantage of cascade connection is that the refrigerants can be operated in their suitable temperature/pressure range. In the low temperature cycle a refrigerant that is especially suited for low temperatures is used and in the high temperature cycle respectively a for high temperature appropriate cooling agent is applied. In the internal heat exchanger, the low temperature refrigerant is condensed and the high temperature refrigerant is respectively vaporized. The net cooling performance is similar to the performance of the low temperature cycle.

2.1.5 Refrigerants

"Cooling agents are working fluids that absorb heat at low temperature and pressure and release it at higher temperature/pressure within a cooling cycle" [8]

Depending on technical boundary conditions and the requirements which need to be satisfied for the specific application, different challenges are exposed on the thermal, thermodynamic and chemical behavior of cooling agents. Apart from operating characteristics also influences on health and environment need to be taken into account.

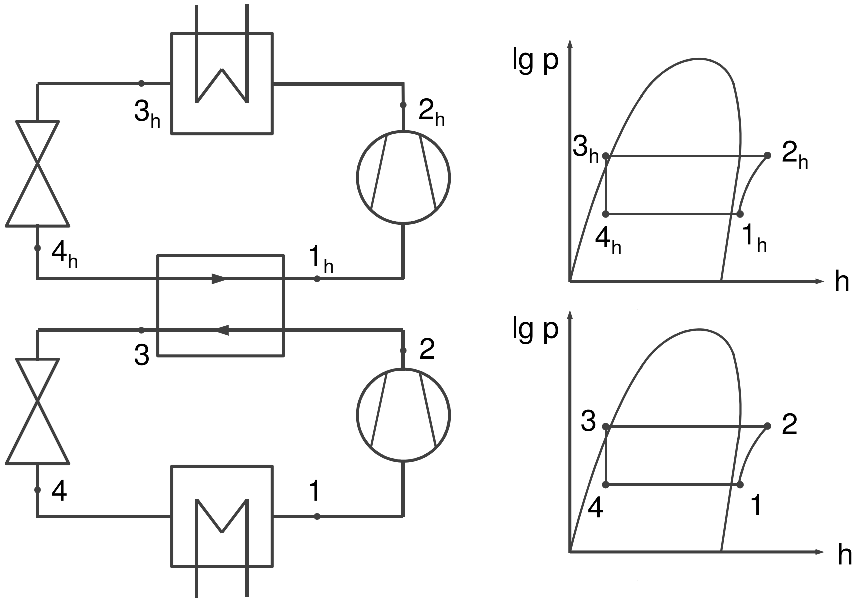

Figure 2.8: Working scheme and log p-h diagram of a two stage cascade connection (own illustration according to [7])

1. Demands on thermal and thermodynamic properties

   An ideal cooling agent should have the highest possible enthalpy of vaporization and a high suction gas density in order to keep the refrigerant mass flow as low as possible. The pressure ratio between vaporization and condensation should be as small as possible in order to minimize electrical power consumption. Furthermore, the critical temperature (triple point) should be well outside the working range.

2. Demands on chemical and physiological properties

   It is important, that the refrigerant has a high chemical stability in order to avoid decomposition at extreme operating conditions. The refrigerant needs to be chemically compatible to the available refrigerator oils as well as construction materials. In addition refrigerants should not be corrosive, flammable and toxic. Cooling agents should be chosen in order to keep their influence on health and security as low as possible. [8]

3. Demands on ecological behavior

   With the consciousness of the global warming and other environmental issues such as the depletion of the ozone layer, the demands on the ecological behavior of cooling agents have increased enormously. For this reason, cooling agents are nowadays assigned to so called ODP and GWP values.

The impact on the environment of certain refrigerants is nowadays often specified by certain measures that are explained in the following.

ODP: The "**O**zone **D**epletion **P**otential" of a refrigerant represents its effect on atmospheric ozone compared to the at the Montreal Protocol[2] determined reference substance trichlorfluormethane which has an ODP value of 1. The atmospheric ozone layer filters ultraviolet radiation which is harmful to the health of human beings.[4] The ozone layer was thinning in the past years, but due to measurements

[2]"The Montreal Protocol on Substances that Deplete the Ozone Layer was designed to limit the production and consumption of the main chemicals causing the destruction of Earth's protective ozone layer. The original Montreal Protocol was agreed on in September 1987 and entered into force on 1 January 1989.[9]

taken over in order to prevent a further depletion, in the last year for the first time a slight recovery of the ozone layer could be observed [10].

GWP: The "**G**lobal **W**arming **P**otential" of a gas is an index that compares the global warming impact of its emission to the environment to an emission of the same amount of carbon dioxide. Usually a lifetime of 100 years is assumed, although the lifetime of carbon dioxide in the atmosphere is way longer. [4]
As those values do not provide a holistic statement on the negative impact of cooling machine to the environment another index TEWI has been introduced.

TEWI: The "**T**otal **E**quivalent **W**arming **I**mpact" considers both direct global warming effects caused by leakages or losses during recycling/maintenance as well as indirect effects meaning the energy consumption and therefore CO_2 emissions which is needed for operation. The common used formula to calculate the TEWI is given by[4]:

$$TEWI = (GWP \cdot L \cdot n) + (GWP \cdot m(1 - \alpha_{recovery})) + (n \cdot E_{annual} \cdot \beta) \tag{2.15}$$

with:

- L being the leakage ratio per year in kg
- n being the system operating time in years
- m being the refrigerant charge in kg
- $\alpha_{recovery}$ being the recycling facor
- E_{annual} being the energy consumption per year in kWh
- β being the CO_2-emission per kWh (energy mix)

Consequently, the first and the second term of the equation represents the direct global warming effects caused through leakage and recovery losses, whilst the third term represents the indirect global warming potential.
Since the negative impacts of former refrigerants had become aware and restrictions have been implemented in the Montreal Protocol in 1987, the use of refrigerants has changed drastically. Before 1987, it seemed that with the chlorofluorocarbons (CFC) adequate cooling agents had been found, as they are non-toxic, non-flammable and equipped with good thermodynamic properties and oil miscibility. Nevertheless they are forbidden now due to the inadmissible high ODP.

Classification of cooling agents

With regards to their chemical composition, refrigerants are divided into two main groups [4]:

1. Inorganic refrigerants

Inorganic refrigerants generally occur in nature and are therefore also often referred to "natural refrigerants". These include water, ammonia, carbon dioxide and generally do not consist out of carbon-hydrogen compounds.

2. Organic refrigerants

Organic refrigerants are usually hydrocarbon compositions. Particularly derivatives of methane and ethane - usually in halogen compounds - are of highest importance in this group. In the past - approximately from 1930 until the Montreal Protocol in 1987 - the chlorofluorocarbons (CFC) together with the hydrochlorofluorocarbons (HCFC) have been the most widely used refrigerants. They satisfied all so far required features such as non-toxicity, non-flammability as well as good thermodynamic properties and oil miscibility. With the awareness of the high ozone depletion potential of those refrigerants, they were forbidden and substituted by hydrofluorocarbons (HFC) which have a negligible small ozone depletion potential but still high global warming potentials.[4]

Blends and Glides Many of the organic HFC refrigerants are mixtures or blends of two or more individual chemicals.[4] Mixtures that exhibit a single boiling point at one particular pressure, having a thermodynamic behavior like single substances are called azeotropes. The boiling temperature of azeotropic mixtures is below the boiling point of the single components. Mixtures that have varying boiling points and compositions throughout the constant pressure boiling process are called zeotropes. Consequently zeotropic mixtures have different compositions in gaseous and liquid phase.

For questions of nomenclature and further explanations please refer to [7],[4],[8].

Current use of refrigerants

In air conditioning applications, for a long time, R22 was the most commonly used refrigerant. Due to the phase out of R22 other refrigerants such as R134a, R407c, R410a became favorable refrigerants to be drawn on for air conditioning applications by manufacturers of compression chillers. R134a (1,1,1,2-tetrafluoroethane) is nowadays widely considered to be the best carbon-hydrogen based single-substance refrigerant. As a result of its beneficial thermodynamic properties it is additionally widely used in refrigerants and car-climate-systems. Since 2011 it is however forbidden to install climate-systems in new cars, that use refrigerants, having a global warming potential that is bigger than 150 [4]. This is to the high risk of leakage in cars and consequently R134a cannot be installed in new cars anyomore. R407C, being the most widely used zeotrope and R410A are both zeotropic mixtures, exhibiting similar properties as R22 and are consequently well suited to replace it in newer applications.
In low temperature applications the zeotropic mixture R404A is frequently used. It has lower critical temperatures and higher pressures than R22 but better heat transfer properties. Compared to other HFCs it has superior low-temperature properties.
Ammonia (R717) has great thermodynamic characteristics and is in comparison to other refrigerants relatively cheap. Disadvantages are the high corrosiveness when using cupper-materials and very high safety-requirements due to its high toxicity. It has been used from the beginning of cooling technology and becomes nowadays more and more important as it neither has an ozone depletion nor a global warming potential. Ammonia mainly finds application in industrial processes rarely in air-conditioning applications.
Another natural refrigerant that is again attracting much interest due to its environmental properties is CO_2. It is being applied in the low stage of cascade systems using the vapor compression cycle. CO_2 is mainly interesting in transcritical processes due to its thermodynamic behavior. Disadvantageous are thereby the high involved pressures and requirements on construction. [4]
Chapter 2.2.3 introduces the currently most important refrigerants respectively working fluid pairs of absorption chillers.

2.1.6 Operational behavior

Compression machines exist in highly diverse designs, so that a universally valid operational behavior cannot be carved out. The behavior is dependent on factors such as layout of the cooling cycles, cooling temperatures, refrigerant, compressor type, etc. Nevertheless some basic operational characteristics that are typically valid for compression machines shall be introduced in the following.
Condenser and evaporator of compression machines are usually operated under relatively constant pressure and temperature levels. The internal performance adjustment is usually refrigerant mass-flow controlled. Compression chillers can handle sudden load changes pretty well due to their excellent dynamic behavior. Nevertheless partial load operation influences the COP of the machines heavily.

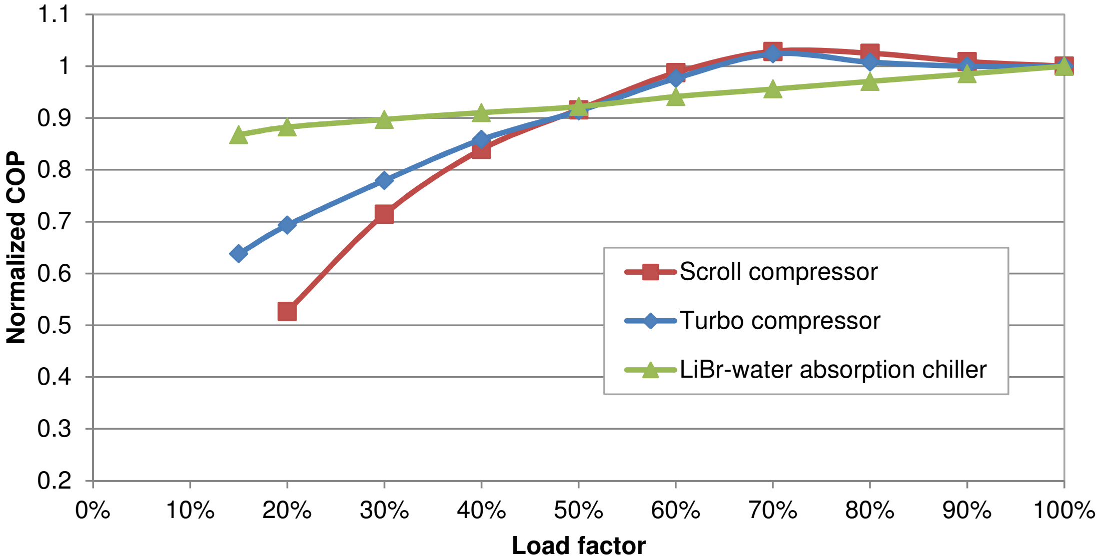

Figure 2.9: Typical partial load behaviour of different cooling machines, normalized to rated COPs. (after [11])

Table2.1 shows typical rated COPs of compression chillers in dependency of compressor type and operating conditions. High cooling performances are partly implemented by screw but mainly by turbo compressors. The heat rejection of the systems can be either dry or wet, whereby for wet heat sink systems usually higher COPs can be reached due to the significantly lower temperatures that can be provided in the condenser. The highest COPs are observed in large machines with turbo compressors. [11]

Table 2.1: Coefficients of performance in dependency of compressor type and operating conditions (own table values from [11])

$T_{recool,in}/T_{recool,out}$ [°C]	$T_{cold,in}/T_{cold,out}$ [°C]	$COP_{rated}[-]$ $Piston-scroll$ $compressor$ $10-1500kW$	$COP_{rated}[-]$ $Screw$ $compressor$ $200-2000kW$	$COP_{rated}[-]$ $Turbo$ $compressor$ $500-8000kW$
water 27/32	6 / 0 14/8	3.6 - 4.1 4.2 - 4.8	4.2 - 4.6 4.9 - 5.4	5.1 - 5.2 5.7 - 5.9
water 40/45	6/0 14/8	2.8 - 3.2 3.3 - 3.8	2.7 - 3.1 3.3 - 3.7	4.1 4.7 - 4.8
air -/-	6/0 14/8	2.4-2.9 3.1 - 3.6	2.7 - 3.2 3.4 - 3.9	- -
pressure ratio of a single stage	-	5 - 10	25 - 30	3.5 - 4
dry heat rejection	-	yes	yes	no
wet heat rejection	-	yes	yes	yes

2.2 Absorption chiller

In contrast to compression refrigeration systems, absorption chillers mainly use thermal energy in order to compress the cooling agent. The working scheme of an absorption machine can be divided into two parts/cycles. The first part is the regular cooling agent cycle, being comprised of a condenser, an expansion valve and an evaporator and thereby corresponds to a compression machine cycle except of the missing compressor. The second cycle substitutes the mechanical compressor of a compression machine. In its basic form it is comprised of an absorber, a solution pump, a generator and an expansion valve. In this second cycle an absorbent with varying cooling agent concentration circulates.

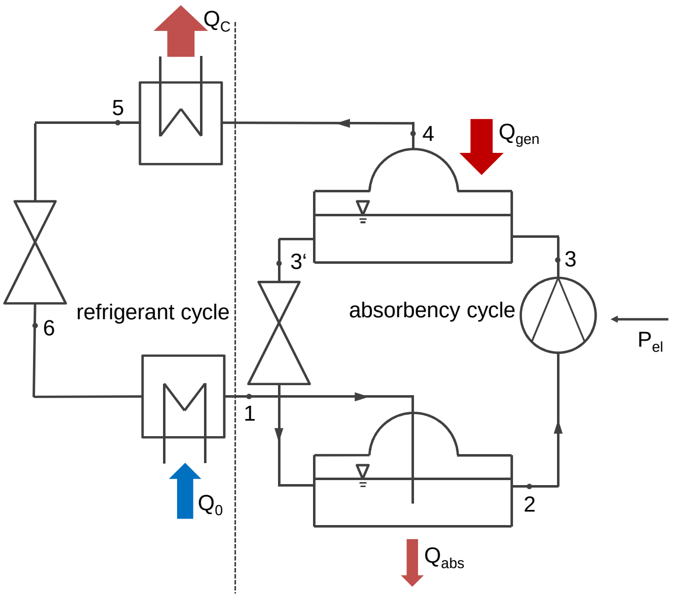

Figure 2.10: Working scheme of an absorption chiller

2.2.1 Absorbency cycle

The method of thermal compression is based on the thermodynamics of binary mixtures. It uses the effect, that the physical solvability of two substances within each other is dependent on the temperature. In the absorber, cold refrigerant vapor meets the refrigerant-weak solution at low temperature. Thereby, the solution absorbs the vapor and its absorbency mass fraction (x) decreases by

$$\Delta x = x_{rich} - x_{poor} \tag{2.16}$$

This process happens under the dissipation of heat, consequently heat has to be removed (Q_{abs}). In technical applications this removal is taken over by a secondary cooling water cycle. The solution pump lifts the refrigerant-rich solution to the higher condenser pressure. The required power for the solution pump is thereby significantly smaller than the power that is needed to compress refrigerant vapor in the compressor of a compression chiller. In the generator, the refrigerant is desorbed from the absorbent by the supply of heat (Q_{gen}). Thereby the refrigerant is vaporized again while the absorbency preferably remains in liquid state if the volatility of the two fluids is in different ranges. If the absorbency has a similar volatility as the refrigerant, a rectifier is included after the desorbing process. The refrigerant vapor is then fed into the "regular" cooling cycle, being condensed, expanded and vaporized again. The refrigerant-poor solution leaves the desorber, is expanded and flows again into the absorber to absorb refrigerant vapor.[11]

Exergy flow

In contrast to compression machines, to which in form of mechanical work, pure exergy is supplied, the heat flow that is running the absorption chiller is tainted by the restrictions of the second law of thermodynamics. In order to produce cooling power, just the temperature dependent exergetic share of the heat flow's energy[3] can be used. For this reason, the overall supplied energy to absorption chillers is higher (cf. figure 2.11) and consequently absorption chillers have increased technical expenditures and require higher heat rejection capacities in comparison to compression chillers.

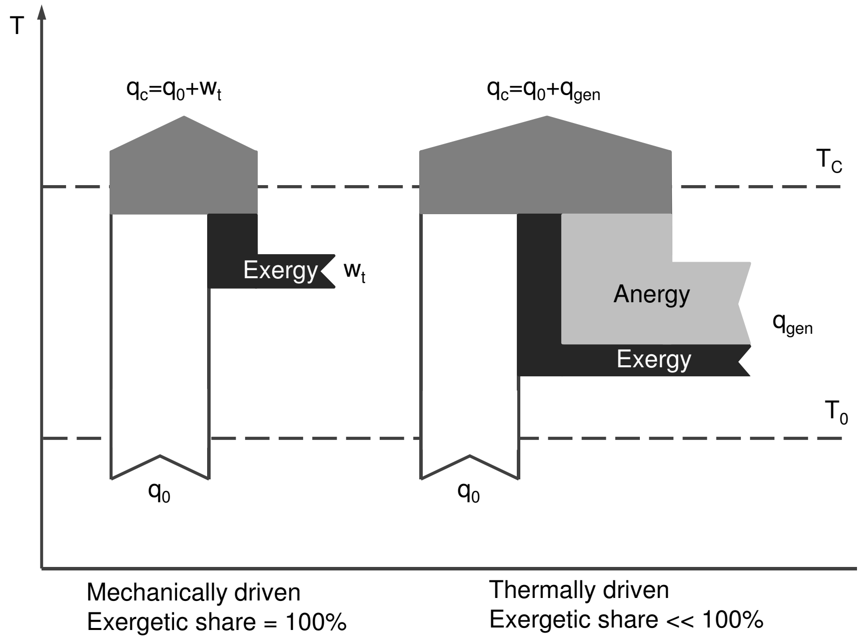

Figure 2.11: Energy flows in cooling machines (illustration according to [12], [11])

2.2.2 Carnot efficiency

The relevant parameter for the evaluation of the efficiency of an absorption chiller is the heat ratio which describes the ratio between absorbed heat Q_0 in the evaporator and supplied heat in the generator Q_{gen}. This ratio is also called coefficient of performance.

$$COP = \frac{Q_0}{Q_{gen}} \tag{2.17}$$

The carnot heat ratio of an absorption machine can be calculated to (for derivation see [2]):

$$COP_{CarnotAM} = \frac{T_0}{T_C - T_0} \cdot \frac{T_H - T_C}{T_H} \tag{2.18}$$

$$COP_{CarnotAM} = COP_{carnotCM}(T_0, T_C) \cdot \frac{T_H - T_C}{T_H} \tag{2.19}$$

[3]The share of the heat flow's energy that can be used for work after thermodynamic equalisation with the respective system[6].

It can be seen, that the carnot efficiency of an absorption machine is additionally tainted by the carnot-factor between heat source and ambient temperature. This is because the absorption process consists of an additional clockwise operated heating cycle (see figure 2.12). In a compression chiller, this factor is already included in the transformation process of heat into electricity in the power plant and thus is not considered in the compression chiller process anymore. For this reason, the COP of absorption refrigerators is by nature lower than the COP of compression chillers. Furthermore formula 2.18 and 2.19 emphasize, that high heat source temperatures and low heat sink temperatures are beneficial for an efficient operation.

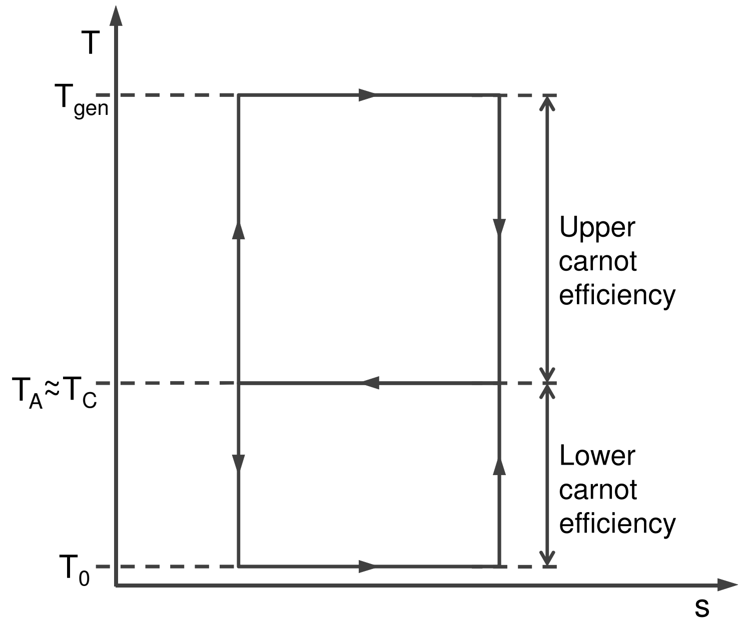

Figure 2.12: Temperature levels in an absorption process - carnot efficiencies

2.2.3 Working fluids

The working fluid-pair for absorption cooling machines is in most cases determined by the selected temperature range and the technical feasibility of the pressure levels. The requirements for useful fluid combinations are:

- The absorbent should have a vapor pressure that is as low as possible in comparison to the refrigerant.
- The heat capacity of the refrigerant and the absorbent divided by the latent heat of the refrigerant should be as low as possible. [13]
- Additives which ensure a better heat/and or mass transfer should not hamper other characteristics such as corrosiveness, thermal stability, toxicity and vice versa.[13]

So far, in absorption technology just two working pairs have established as a practicable fluid combination, fulfilling these requirements best.

Water/lithium bromide solution

The first fluid combination uses water - as cooling agent and lithium bromide solution - as absorbent. The advantages of water as a refrigerant are numerous. Water has a good enthalpy of vaporization, a low vapor pressure - which enables small wall thicknesses. Both fluids neither have an ozone depletion potential nor a global warming potential, are neither toxic nor flammable. Another advantage is that the vapor pressures of the two fluids are very different. The lithium bromide solution is considerably less volatile than water and therefore makes a rectifier following the desorber unnecessary. However by the use of the working pair water/lithium bromide, the cooling temperature is restricted. It is not possible to undergo the freezing point of water and therefore in reality the lower limit of cooling operations is in the magnitude of +4/5°C. A further disadvantage is the narrow solution field, that is limited by the solvability of lithium bromide salt in water and can cause crystallization (see figure 2.14). In addition the fluid combination imposes high requirements on the leakproofness of the machine. This is because extremely low vacuums are necessary for the vaporization at low temperatures. [7][13]

Ammonia/water

The second fluid combination is mainly used in applications in which lower temperatures are required and for which water/lithium bromide cannot be used anymore due to the mentioned restrictions. Ammonia also has a high enthalpy of vaporization and is applicable until cooling temperatures of about -60°C. It involves good heat- and material transmission conditions and standard steals can be used. Compared to water in lithium bromide, ammonia has a high solvability in water and therefore a wider solution field. However ammonia also implies disadvantages such as a high toxicity and a rather high vapor pressure of the cooling agent, making big wall thicknesses necessary. Another disadvantage is, that the volatility of ammonia and water is in similar ranges and consequently a downstream rectification is required. [7] The rectification declines the COP as additional heat is required for this process.

Research - Example Ionic liquids

Even so far no other fluid combination could establish itself in market, there is still research going on in respect of finding new efficient and safe fluid combinations. One promising solution could be the use of the new fluid-class "ionic liquids" as absorbency. Ionic liquids are organic salts that are liquid at room temperature and their properties can be selectively determined by the choice of anions and cations. Organic salts in combination with refrigerants such as water form a promising combination as working fluid pair for absorption chiller technology. Advantages are:

- no toxicity, no explosiveness
- high chemical resistance until 150°C
- high solubility of water
- low vapor pressure (water)
- totally miscible with water in the temperature interval of interest

Promising investigations about this working fluid combination are going on. Nevertheless there are still obstacles which have to be passed before the fluid-combination can reach market maturity. In comparison with lithium-bromide water absorption chillers, ionic liquids have worse properties regarding heat transfer. Therefore further research regarding the geometrics of absorber and heat exchangers need to be conducted. However, ionic liquids could open new possibilities regarding construction materials as they are advantageously with respect to corrosiveness over lithium bromide machines. [14]

2.2.4 Absorption chiller concepts

Each manufacturer has its unique special layout underlying its absorption chillers. Nevertheless, certain basic features and basic layouts that are similar for most machines can be observed and are presented here. Depending on the specific case and boundary conditions of the project, different working schemes are relevant. Factors which are taken into consideration are on the one hand the technical efficiency and on the other hand investment costs and consequently the profitability of the system.

Single stage chillers The least sophisticated modification, which is however preferably used, is the single stage absorption chiller. Its basic working scheme has been already explained and shown in figure 2.10. Single stage means in this context, that the system simply consists of one absorber and one generator. Almost every single stage machine that is available on market has integrated an additional heat exchanger between the refrigerant-rich absorbency solution at the exit of the absorber and the low concentrated refrigerant/absorbency solution in order to increase heat utilization and therefore improve efficiency (see figure 4.1). In case of the working fluid combination ammonia/water, a rectification column is arranged following the generator in order to receive highly pure ammonia vapor at the inlet of the condenser in the refrigerant cycle.
However single stage processes have certain restrictions regarding temperatures that cannot be crossed. When the temperature of the heat rejection circuit is too high or the heat source temperature is too low, the required evaporator temperature[7] cannot be provided. In figure 2.13 marginal conditions are exemplarily given for ammonia single stage absorption chillers. The diagram is based on a degassing width Δx of 6%.

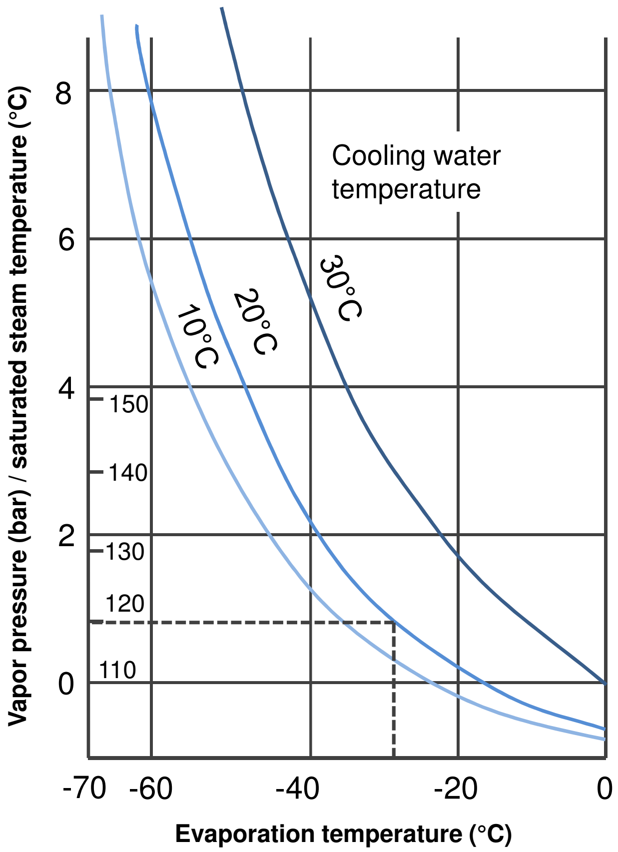

Figure 2.13: Marginal conditions for single stage absorption chillers (illustration after [7])

In the diagram the minimum required heat source temperature for given evaporator and cooling water temperatures respectively the maximum evaporator temperature for given heat source and cooling water temperatures can be read off. For example for cooling water temperatures of 20°C and evaporator temperatures of about -30°C, a heat source temperature of minimum 120°C is required. Furthermore the possible temperature spreading between cold water, cooling water and hot water is limited by the solution field of the respective fluid combination. This is particularly of importance for machines that use the fluid combination LiBr/water. Figure 2.14 shows the solution field of this fluid combination in a so called Dühring plot. In this plot, the abscissa of the diagram shows the boiling point of the solution, whilst the ordinate shows the saturation temperature of the refrigerant and the vapor pressure. The saturated liquid lines of constant concentration can almost be considered linear. Furthermore the equilibrium state points of an exemplary single stage absorption process (absorber A, generator G, condenser C and vaporizer V) are inserted in figure 2.14. It can be seen, that for an evaporation temperature of 5°C, a condenser temperature of 45°C and a relatively small degassing width of 4 %, already a heat source temperature of nearly 100°C is required to run the process successfully. If the condenser temperature would be increased to 55°C, an operation of the process with a degasing width of 4% would not be possible as the limit of solubility (red line) would be exceeded and therefore lead to crystallization of LiBr.[15]

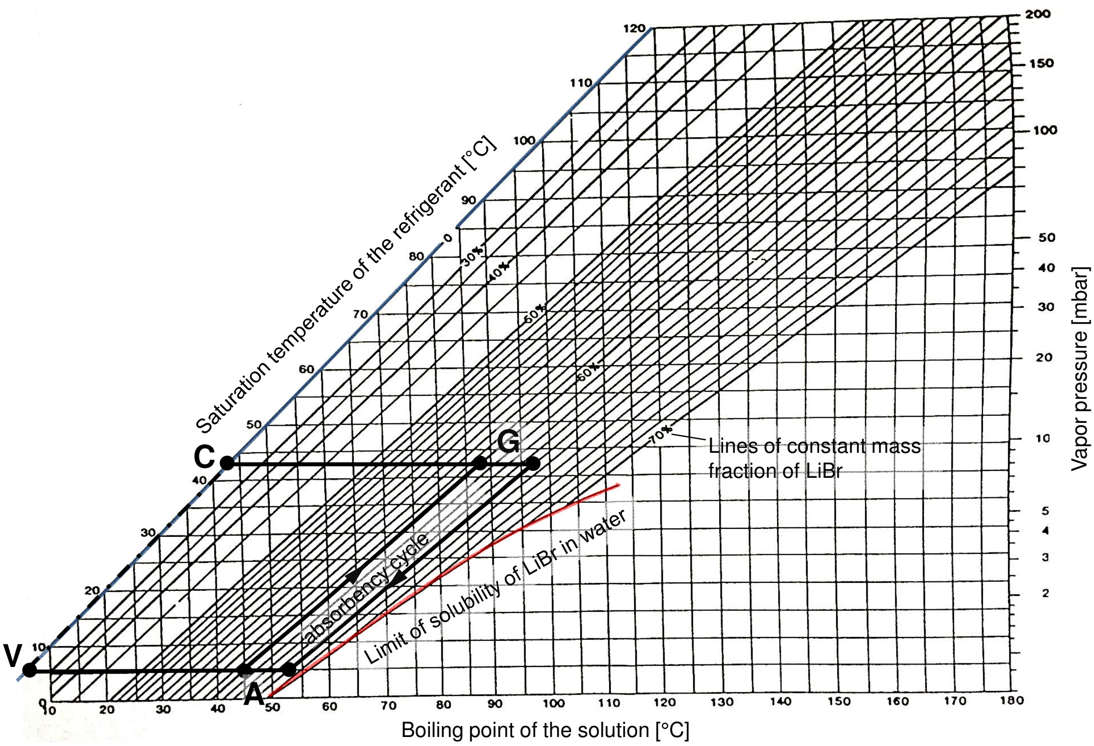

Figure 2.14: Solution field for LiBr-water (modified Dühring plot from [13], [15])

The *COP* of absorption chillers is not directly linked to the temperature difference between evaporator and condenser. If this difference is increased, there is the possibility to compensate by an increase of the heat source temperature and only if this is not possible, the heat ratio is lowered in a second step. A disadvantage is however, that the heat ratio of single stage absorption chillers for this reason can never exceed the value 1. By the use of multiple stage cycles, this limitation can be overcome. [15]

Multiple-stage/effect absorption chillers

Depending on the specific boundary conditions of a project, different interconnections with diverse features and qualities can be chosen. It exist myriads of different interconnections and in the following some

important and commonly used interconnections are presented. For a holistic view over the different cycle possibilities of absorption machines, it is however referred to the works of [15] and [13].
The chosen way of presentation of the different absorption cycles is correspondent to the works of [13] and [15]. Multiple stage absorbers are depicted by the connection of two or more elementary absorption cooling cycles in p-T diagrams (cf. figure2.14).
Circles at the corner of the cycles represent heat transfer units (evaporator, absorber, desorber, condenser). Curved lines describe an internal heat transfer between inlet and outlet of the respective exchange units. Diagonal lines represent the vapor pressure line of the refrigerant (in case of a connection between condenser and evaporator) and the saturated liquid line of the absorbent (in case of a connection between absorber and desorber). [15]
A basic distinction is made between multiple-effect and multiple-lift absorption cycles, both describing the number of stages of the machine.

High efficiency interconnections - multiple-effect absorption machines

When heat at high temperature level is available, multiple-effect cycles can often be used in order to increase system performance. In multi-effect cycles, the system is configured in a way that heat, which is rejected from a high temperature stage/cycle, is supplied to a low-temperature stage for the generation of an additional cooling effect. In simplistic terms, by the supply of one unit of heat, two units of cold are produced. In reality, the COPs of the individual cycles within a multiple-effect interconnection are not exactly twice as high as the COP of a single stage cycle would be. Nevertheless, the overall COP of those systems exceed single-stage systems often remarkably. [16][17].

Double-effect cycles

In figure 2.15, two double-effect cycles are shown. Interconnection 2.15a is operated at three pressure levels while connection 2.15b is run on only two pressure levels. In general, devices with interconnection 2.15a involve higher pressures than devices operated by interconnection 2.15b, which is a disadvantage for the use of ammonia, as really high pressures are reached (usually $\geq$ 80bar). For this reason, this interconnection is more suitable for the use of LiBr-water and therefore often used and commercially available in double-effect absorption machines using this fluid combination. Configuration 2.15b requires a wider solution field than 2.15a and is therefore mainly applied in water-ammonia absorption machines. The pressures that are reached are usually significantly lower and do usually not exceed 25 bar [13]. In interconnection 2.15a the rejection heat of the high pressure condenser is used to run the low pressure/temperature desorber. The cycle of 2.15b is designed in a way, that the low temperature desorber can be run by the absorption heat of the high temperature absorber.
The application field of double-effect absorption chillers usually starts when heat of at least 120°C is available. Typical values for the COP are then in a range of 0.8-1.2. [13] [16]

Triple-effect cycles

As the name already indicates, triple-effect cycles are capable of tripling the heat ratio in comparison to a single-stage cycle. Figure 2.16 shows three triple-effect cycles. While cycle 2.16a is comprised out of seven exchange units, interconnection 2.16b and 2.16c are composed out of 8 exchange units. In cycle 2.16a, the absorption heat and condensation heat of a first cycle are used to drive a second cycle. In cycle 2.16b, the high pressure condensation heat of an additional cycle is used as driving energy for a double effect unit. The same applies to 2.16c - however at 4 pressure levels. Consequently

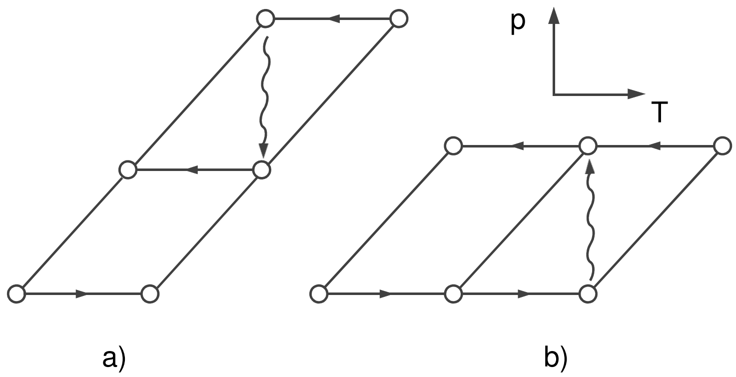

Figure 2.15: a) double-effect cycle with 3 pressure levels; b) double-effect cycle with 2 pressure levels

this cycle causes high pressures and hence is - if at all - applicable in LiBr-water machines. In order to make triple-effect units profitable/reasonable, very high heat source temperatures of at least 200 °C are necessary. As a result of those high temperatures, triple-effect machines are likely to be direct fired. The high operating temperatures lead to bigger corrosiveness of the solution and therefore increase maintenance expenditures. COPs of triple-effect machines are usually in the range between 1.4 and 1.5.[16] The higher number of effects leads to increased system complexities. For this reason, triple- and quadruple-effects are rarely available and double-effect systems are the only commercialized multiple-effect machines. [16]

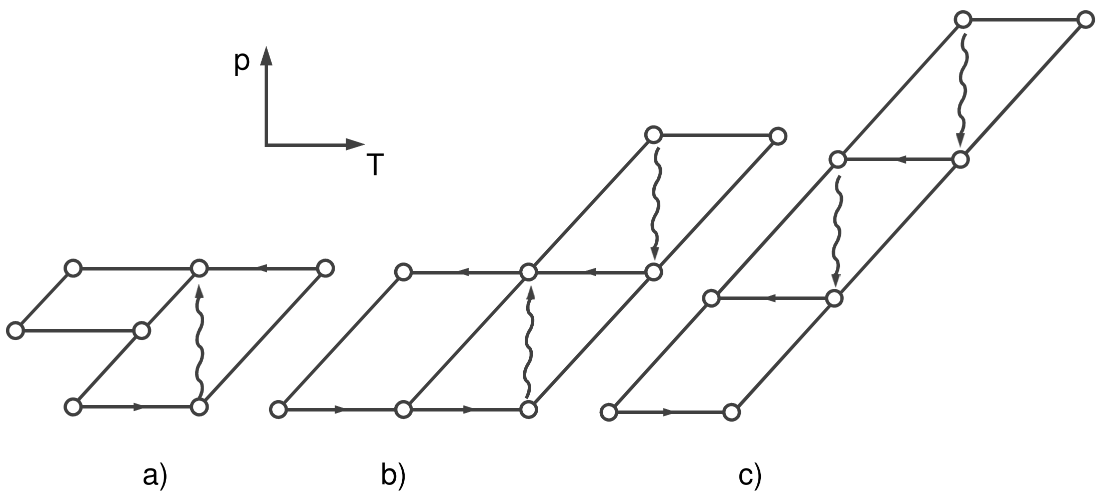

Figure 2.16: Triple-effect interconnections

Large temperature lifts - double-lift absorption chillers

Multiple-lift absorption chillers raise interest if the available heat source temperature is too low in order to provide the required evaporator temperature in a single stage process. By the help of multiple-lift cycles, the cooling agent vapor can be transferred to condenser pressure in two - or more stages. The term "lift" in this context describes the single or repeated supply of heating energy. In simplistic terms, in a double-lift machine, two units of heating energy are necessary in order to generate one unit of cold. For this reason, the interconnections do not increase the COP of the process - rather lower it - but make use of heat at lower temperatures. The typical single-effect, double-lift interconnection is depicted in figure 2.17. At intermediate pressure, refrigerant vapor is desorbed in a first stage. In the following it is absorbed again at intermediate pressure and lifted to the high pressure. There it is desorbed again by further heat supply. Typical COPs of double-lift chillers are in the range of 0.3-0.4. [16]

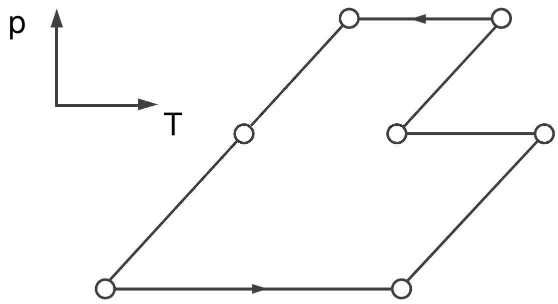

Figure 2.17: Double-lift interconnection

2.2.5 Operational behaviour

Real operating parameters often differ from laid-out conditions and performances. This is due to partial load operation and different disturbances such as poorly heat rejection due to extreme ambient conditions. The temperature at cold water side is in general kept constant, as a certain cooling temperature should be guaranteed. The same applies to the heat rejection cycle inlet temperature that is usually kept constant. A performance adjustment is therefore usually regulated by either the heat source temperature - in case of a constant volume flow in the generator - or by a variable volume flow - if the heat source temperature is kept constant. For the reason that the volume flows of cold water and cooling water are usually kept constant, the compensation for the different cold performances is done by variable outlet temperatures of cold water and cooling water.[11]

In figure 2.18 the temperatures of inlet and outlet of hot water, cooling water and cold water can be seen for an exemplary absorption machine. The mentioned effects can be well observed. The inlet temperature of the cold water is relatively constant at 5°C. The cold water outlet temperature is however reduced to about 7.5°C for small performances and increased to 12°C for maximum performance. Similar behaviour is observed for the cooling water. The inlet temperature is constant, given that the heat sink system can provide the necessary temperature over the entire performance range. The cooling water returns to the heat sink system however with higher temperatures for increasing cooling performance. This can be lead back to the constant volume flows of cold water and cooling water. For the hot water a relatively constant temperature difference between inlet and outlet temperature can be observed. In this exemplary case, the cold performance is regulated by the heat source temperature.

It can be seen that fluctuations in temperature increase with decreasing load factor. Nevertheless absorption chillers have an avowed good partial load performance (cf. figure 2.9). The COP typically differs only slightly within the load range which is usually down to 10%-30% of the rated power. As soon as the minimum load is however undergone (in this case 400kW), fluctuations become so immense, that a controlled operation is not possible anymore.

2.3 Adsorption chiller

Sorption machines that use solid substances as sorbent are called adsorption chillers. The general thermodynamic working principle is the same as for absorption chillers. The technical implementation differs however, as a solid sorbent cannot circulate. For this reason, adsorption processes are in contrast to absorption processes discontinuous processes and divided into two working phases. In a first phase,

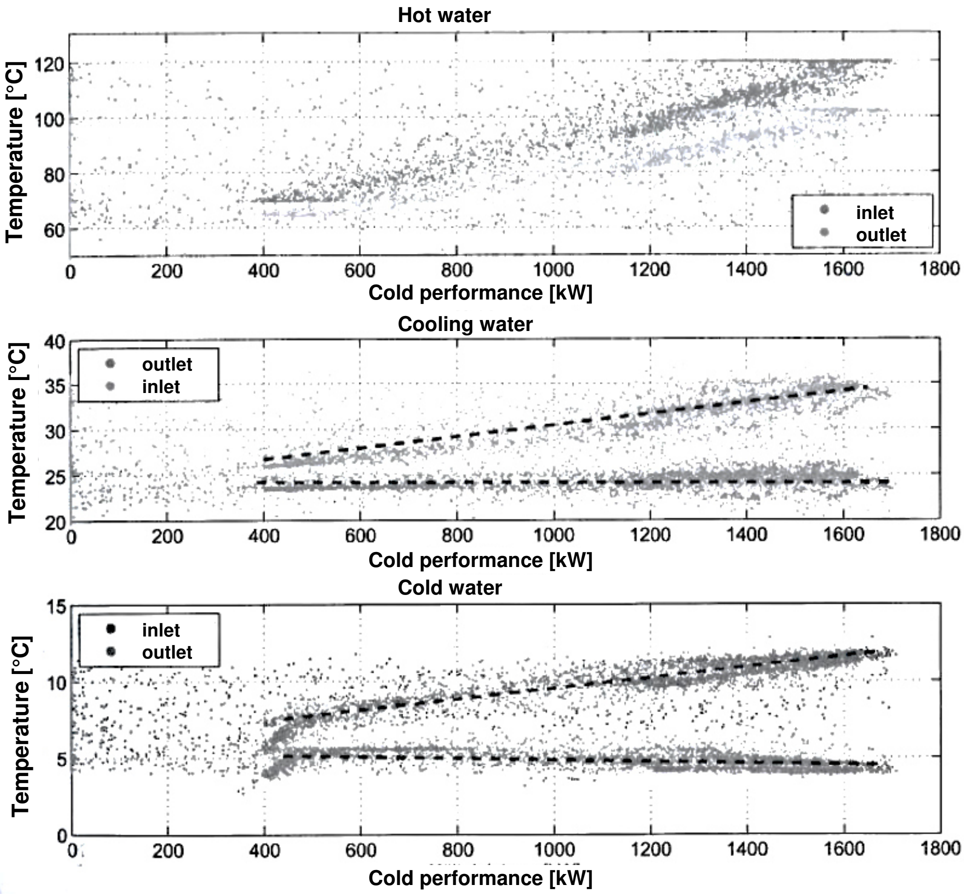

Figure 2.18: Temperatures in dependency of the cooling performance. Real hourly average values of an absorption chiller.[11]

the cooling agent is vaporized by the intake of heat in the evaporator and the vapor is adsorbed by the adsorbens under dissipation of heat. The second phase includes the desorption of the refrigerant by the supply of heat. Like in an absorption process, the sorption ability of an adsorbent decreases with increasing temperature. In the condenser, the refrigerant vapor is then condensed and heat is released. It is always possible to intercept the adsorption process for a certain time by the use of valves. For this reason, adsorption machines can also be used as cold or heat storage. Furthermore it is possible to reach nearly continuous operating conditions by interlinking two or more adsorption chillers that are operated in opposing phase. In figure 2.19 both phases of the process are shown. In phase 1, the adsorber is connected to the evaporator and the desorber to the condenser. When the phase is shifted, the cross valves switch position, so that the adsorber becomes the desorber and is connected to the condenser and vice versa (phase 2).[18] Currently the only market-based substance combinations are water as refrigerant and zeolites or silicagel as adsorbent. [19] Adsorption machines are inferior to absorption chillers with regard to its size and specific performance. Further problems are represented by the preservation of a sufficient sealing in order to maintain a vacuum and the problems that come along with the cyclical operation mode (fluctuations in temperature and energy flows). Advantages in comparison to absorption chillers lie in the harmlessness of the working fluids and materials, the better solvability without potential danger of crystallization (in comparison to LiBr machines) and a minor maintenance effort. Furthermore it has a better behavior regarding the operation with low temperature heat sources ($\geq 55°C$). For this reason adsorption machines are e.g. in combination with solar thermal systems a promising technology. In summary, adsorption chillers are a promising technology for smaller applications with low heat source temperatures but not yet for bigger applications that would be suited

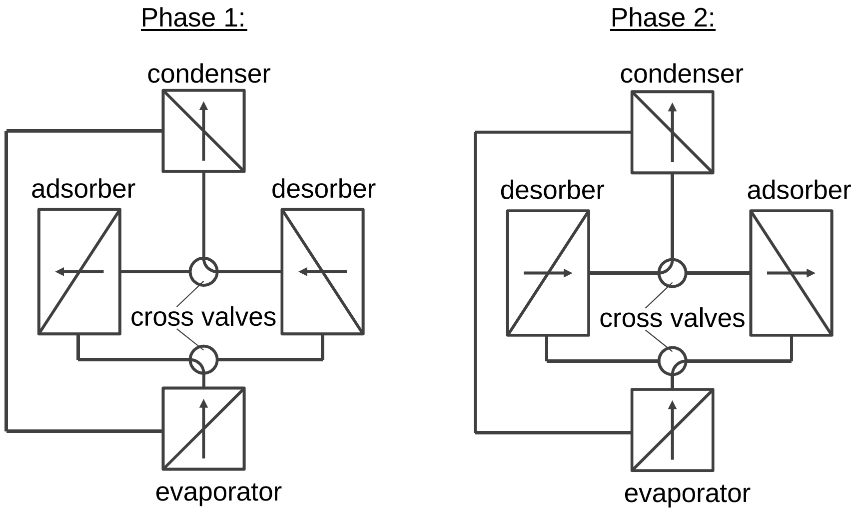

Figure 2.19: Working scheme of an adsorption chiller depicted in 2 different working phases [18]

for district heat. On that account and the low market introduction, adsorption chillers are not element of the subsequent considerations.[11]

2.4 Steam-jet chiller

The working scheme of a steam-jet chiller also displays similarities to that of a compression machine. As well as in absorption/adsorption machines, the mechanical compressor is substituted by a thermal driven component, a steam-jet compressor. The process is operated in two cycles. The first one is the classical cooling cycle in which the refrigerant circulates. The second one is the cycle of the propellant. Usually water is both used as refrigerant and as propellant.

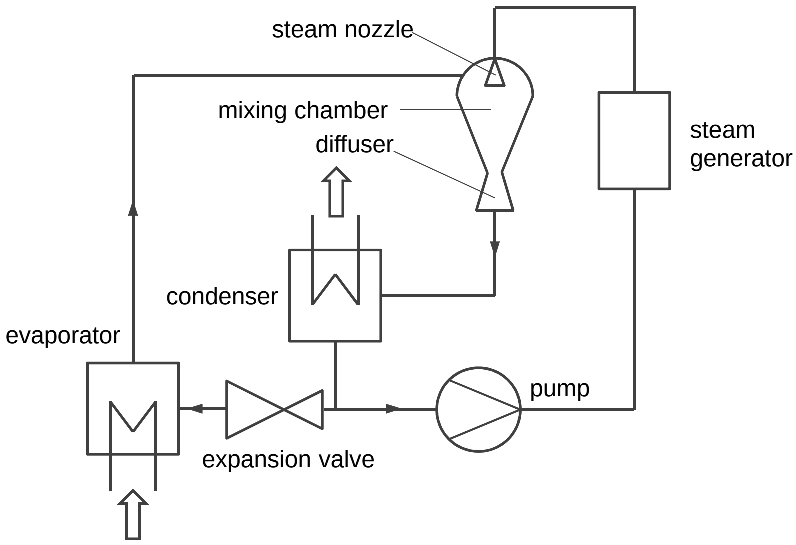

Figure 2.20: Working scheme of a steam-jet chiller (after [2])

A steam-jet compressor basically consists of a steam nozzle, a diffuser and a mixing chamber. Via the steam nozzle, the potential energy that is proportional to the respective pressure at the inlet of the nozzle is converted into kinetic energy. The static pressure of the steam flow is thereby lowered below evaporator pressure, so that vapor is sucked from the vaporizer into the mixing chamber. The vapor mixture that is emerged in the mixing chamber enters then with high velocity - usually higher than supersonic velocity - the diffusor. In the diffusor the process is operated vice versa and the kinetic energy is transferred again into potential energy (high pressure). The increase in pressure thereby has to be high enough in order to liquefy the entire vapor with the given cooling water. Through the expansion valve, a small part of the fluid is fed back into the evaporator, while the bigger part is pumped into the steam generator. [2] In contrast to compression chillers, big steam volume flows are not critical so that water can be used in the interesting temperature field within 0-10°C. Steam-jet chillers are of particular interest, when low-pressure vapor is cheaply available. The low constructive effort, low susceptibility [2] and COPs in the range of 0.2-1.2 depending on heat source temperature (85°C-180°C) [11] and cold water/cooling water temperatures make steam-jet systems to a technology, worth considering in the future. However within the scope of this work this technology is not further considered due to its currently narrow assortment and low market maturity.

2.5 Additional components of cooling systems

2.5.1 Heat sink systems

A very important part for the evaluation and the calculation of the costs of a cooling system is the heat sink system. Each cold vapor machine needs to release heat out of the system in order to maintain a stationary cooling cycle. The energy and consumables such as water that are needed to run the heat sink system are not considered in the calculations of the COP of cooling machines. However those factors influence the overall efficiency of the cooling system considerably. Usually, heat is not released directly to the environment but through an intermediary closed fluid-cycle (see figure 2.21). The heat of the condenser is released to the closed heat rejection cycle which in turn removes the heat to the environment. For this removal, several different working principles exist. In general a distinction is thereby made between convection based and humidification based as well as a combination between both - hybrid systems. In the following, the most common and important heat sink types as well as their drawbacks and benefits are briefly explained. [7]

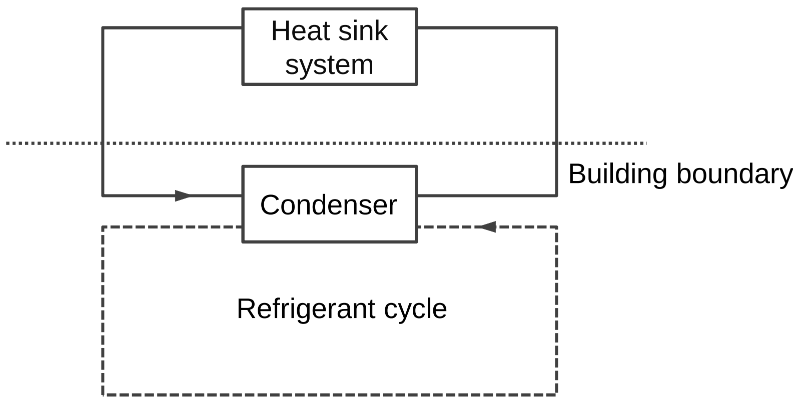

Figure 2.21: Embedding of heat rejection systems into cold generation systems

Dry cooler

A dry cooler consist in general of a finned heat exchanger which dissipates the heat to the ambient air by means of natural or forced convection. Within the pipes of the finned heat exchanger the cooling fluid is flowing (typically water or a water-glycol-mixture). In a dry cooler, cooling temperatures that are slightly higher than ambient temperature can be reached. The temperature difference between cooling fluid and ambient air is usually in the range of 4-5K and results from heat transfer reasons. With respect to sonic emissions of fans in the case of forced convection, dry coolers are often limited regarding the installation site in order to comply with the sonic noise limits.

Evaporative Cooler

Open evaporative cooler / cooling tower An open evaporative cooler - also often called cooling tower - consists of a housing in which a widespread packing is included. Above the packaging a nozzle holder is situated. Through this nozzle holder, the water that is to be cooled is sprayed into the packaging. The water percolates through the packaging and at the same time air is sucked in the opposite direction through the packaging by fans. Thereby, a certain amount of water vaporizes and absorbs heat, but the majority of the water is cooled down and flows into a collecting trough which is situated at the bottom of the packaging. During the process, water is consumed by evaporation, desludging and splash water. These losses must be compensated by the addition of fresh water. Furthermore, water treatment facilities such as water purification plants, water filtration systems and water softening systems are required in order to guarantee faultless functioning. [4][7]

Closed evaporative cooler The basic construction of the closed evaporative cooler is the same as for the open version. The difference is however that the fluid that is to be cooled is flowing through a plain tube and is not sprayed into a packaging. Consequently, there is no direct contact between the fluid and the air that is blown by pumps through the housing. The water that is collected in the trough at the bottom of the housing is pumped above the plain tubes and sprayed on it. By the evaporation of the water on the pipe surface, the cooling effect is realized. The mass flow of water over the condenser tubes must be high enough in order to guarantee the wetting of the tube surface and is usually 80-160 times the quantity that is evaporated. The mass flow of air must be sufficient to carry away the water vapor. [4][7]

Hybrid dry coolers

Hybrid dry coolers are a mixture between dry coolers and cooling towers. At low ambient air temperatures, heat rejection is achieved by convective heat transfer to the ambient air. At higher ambient air temperatures, it is possible to moisten the surface of the air heat exchanger so that heat is transferred partly convective and partly latent by the vaporization of water. The higher the surrounding temperature, the higher is the rate of latent heat transfer. At laid- out conditions, meaning maximum ambient air conditions, the heat transfer occurs almost 100% latent. For this reason, the peak water consumption rate is in a similar range as for pure evaporative coolers. Hybrid cooling towers however raise interest if the ambient air temperature and/or the performance drops and therefore the water consumption is drastically reduced. For ambient air temperatures $\leq$ ca. 12°C, the cooling is usually done solely dry. If the cooling capacity is reduced, dry operation can usually be expanded to ambient air temperatures of about 15-18°C. [7]

Comparison

For the efficiency of the cooling cycle, evaporative heat sink systems represent in general the best option, as it is the technology that enables the lowest cooling water temperatures. On the other hand, wet heat rejection systems imply usually high costs for consumables such as water, which in turn reduce the profitability of the system. Furthermore, the risk of the occurance of legionella that find optimal temperature conditions in the range of 25°C-35°C could not be abandoned completely. An assessment of the system components and site requirements is therefore always done in order to find the right heat sink system.

Table 2.2: Charakteristics of heat sink systems (own table - values from [11]. Advantageous (+) and disadvantageous (-))

	Dry cooler	Open evap. cooler	Closed evap. cooler	Hybrid cooler
Noise emission	- -	+	+	-
Dependency on ambient conditions	- -	+ +	+	0
Water consumption	+ +	- -	- -	-
Thermodynamic evaluation	-	+	+	+
Required space (size)	-	+	+	+
Legionella	+ +	- -	- -	-

2.5.2 Buffer storage

Buffer storages often take over an important role for the well-functioning of a cold generation system. By decoupling the volume flows of cold consumer and cold generator, they principally create the prerequisites for the adherence of a rather constant cold water flow temperature during spontaneous load changes and/or possible short-term downtime of the refrigeration system. Changes in temperature are usually displayed as disturbances of the subsequent control of the chiller leading to increased energy consumption as well as noncompliance of the setpoints and therefore should be prevented.
Furthermore buffer storages have the function to ensure cold supply if the minimal partial load of the chiller is undergone. For this reason, the clock cycle of the chiller is generally the defining parameter for the dimensioning of buffer storages for compression systems. The general rule for those systems is, that the buffer storage must store the minimal partial load for the time of the clock cycle of the machine.
The volume of a cold-water buffer storage V (in l), needed to provide a certain cold capacity Q (in kWh) for a certain time t (in min) can be calculated by the following formula [20]:

$$V = \frac{Q}{c \cdot \Delta T} \cdot \frac{t}{60} \tag{2.20}$$

with:

- c being the specific heat capacity of water, amounting to 1.163 Wh/lK
- $\Delta\, T$ being the temperature difference between cold water inlet and outlet in K
- the equation obtains the consideration that 1kg of water approximately occupies a volume of 1l

For compression systems, equation 2.20 is used for the dimensioning in the praxis but additionally assigned by factors such as a mixing factor f_m, which value is dependent on the type of application and amounts between 1.3 and 2.0.[20] Recommendations for the dimensioning of buffer storages for absorption systems are given in chapter 4.2.
Figure 2.22 shows schematically a cold water buffer and the conventional embedding into a cold generation system. It can be seen that the buffer storage is integrated in the cold water cycle in between of the cold generator and consumer. The storages usually work by the principle of thermal stratification.

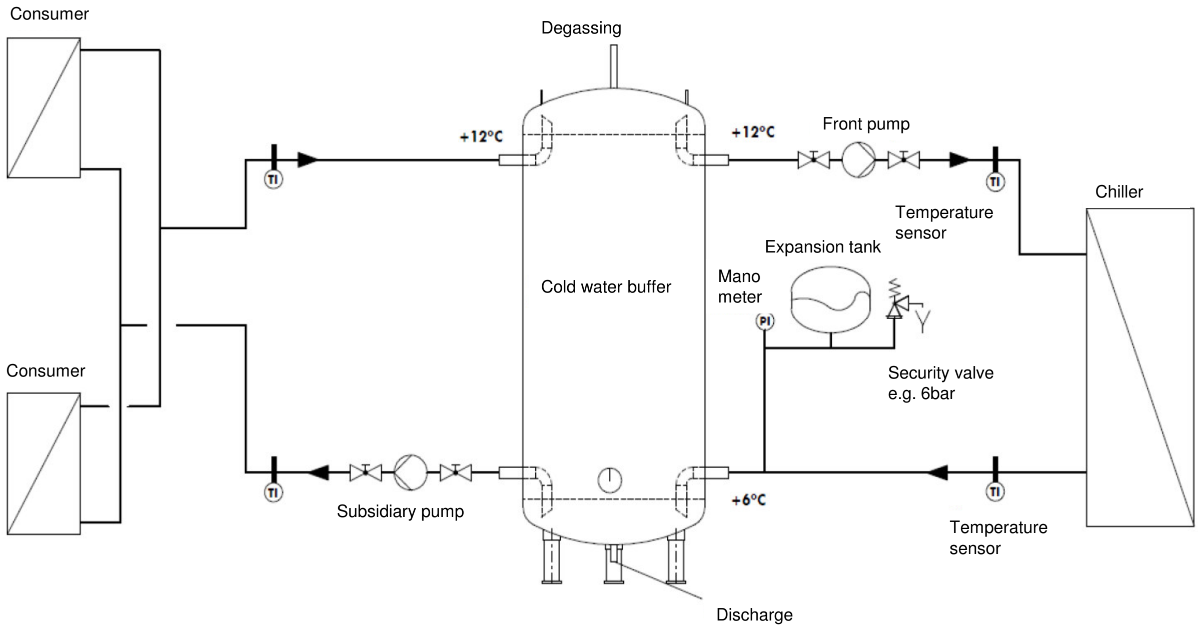

Figure 2.22: Embedding of a cold water buffer into the cold generation system (translated from [21])

2.5.3 Free cooling

When the ambient temperature falls below the required cold water temperature, and there is still a certain cold demand, possibly free cooling is a meaningful supplementation to the cold generation system. In this case, no further energy input to the operation of the refrigeration systems is necessary but heat is removed either directly or indirectly via the secondary cycle of the heat sink system, that is usually run by a water-glycol mixture. [22]. In case of direct free cooling, cold ambient air is directly lead to the heat source via channel systems.
However the more common variant is the indirect free cooling in which the water-glycol circuit of the heat rejection system is connected to the primary cold water circuit via a heat exchanger (usually a plate heat exchanger).
For applications that have a remarkable cold demand in the cold winter months, by the help of free cooling systems, a considerable amount of operational hours can be taken over, the energy consumption lowered and cost savings be reached.

2.6 Summary and comparison

With regard to the suitability of the use of district heat in order to provide cooling power, lithium-bromide water machines seem to be the most mature thermal driven technology in market.
Ammonia-water absorption chillers tend to require higher heat source temperatures than the ones that are typical for district heating systems in summer (75°C - 85°C inlet temperature), particularly when low cold water temperatures are required (cf. figure 2.13). Table 2.3 provides an overview over the properties and temperature ranges of the single stage absorption machines using the different working fluids. The application field is therefore already concentrated to cold water temperatures that are bigger than 5°C due to fluid properties. In a comparative contemplation between absorption and compression chillers, the in table 2.4 listed advantages of compression chillers respectively absorption chillers can be observed. Absorption chillers exhibit higher initial investment costs, (see chapter 6). In order to be competitive to compression chillers, lower operational costs should be able to compensate the deficit within

Table 2.3: Overview: properties and temperature ranges of single-effect absorption machines (data from [11])

	Absorption chiller $NH_3-H_2O(SE)$	Absorption chiller $H_2O-LiBr(SE)$
Operating principle	continuous	continuous
Properties		
Toxicity	high	low
Corrosiveness	high	very high
Flammability	high	-
Temperature stability	until 180°C	120 - 130°C
System pressures	$\geq 20bar$	smaller 1bar
Solubility	good	bad
Necessity of rectification	yes	usually not
COP	0.35 - 0.65	0.55 - 0.8
Application range		
Heat source temperatures	100 - 160°C	75 - 140°C
Cooling water temperatures (inlet)	15 - 45°C	15 - 45°C
Cold water temperatures (outlet)	-50 - 5°C	5 - 25°C

a certain period. For this reason, absorption chillers should be operated when a cheap heat source is available and when the characteristics of the load profiles fit to the advantages of the technology (see chapter 3).

Absorption chillers need significantly less electric energy and use thermal energy as main driving energy. This leads however to the necessity of larger heat rejection capacities, as the thermal energy is in contrast to electric energy not pure exergy. Furthermore absorption chillers require more space for the housing of the system as well as for the placement of the recooling system.

A big advantage of absorption chillers is the use of climate neutral cooling agents and working fluids. Both commercially available combinations neither exhibit a global warming potential nor an ozone depletion potential.

With regard to the partial load behavior, absorption chillers represent the better option - the COP usually does not vary substantially within the given performance range. The partial load behavior of compression chillers is in this comparison significantly worse. Nevertheless they have a monumental better dynamic behavior (clock cycle times between 1.5 and 3 minutes [20], absorption chillers about 15-20 min (indications of manufacturers given in the course of the inquiry)) being able to react on sudden load changes. Absorption machines hereby come up with a rather sluggish behavior, that is why it is recommended to install additional buffer capacity in order to smoothen sudden load changes.

Table 2.4: Comparison between compression chillers and absorption chillers (data partially from [11])

	Compression chiller	**Absorption chiller**
Working principle	Vaporization at low pressure Condensation at high pressure	Vaporization at low pressure Condensation at high pressure
Type of compression	Mechanical compression	Thermal compression
Driving energy	electricity	heat
Declaration of efficiency	COP = Q_0/P_{el}	COP = Q_0/Q_{gen}
Typical efficiencies	3-6	0,6 - 0,8 (single stage,LiBr)
Advantageous (+) / Disadvantageous (-)		
Compact installations	**+**	**-**
Investment costs	**+**	**-**
Electricity consumption	**-**	**+**
Required heat rejection capacity	**+**	**-**
Maintenance expenditures	**-**	**+**
Possibility of dry heat rejection	**+**	**-**
Dynamic behaviour	**+**	**-**
Reliability	**-**	**+**
Partial load behavior	**-**	**+**
Environmental compability of working fluids / refrigerants	**-**	**+**

3 Fields of application

In order to be able to assess the applicability of absorption chiller technology, basic considerations about the fields of application and suitable load profiles need to be performed. In a first step, the available heat source - the district heating grid - and its features and restrictions are shortly introduced. Secondly, conditions, that are advantageous for absorption chillers, are derived from the characteristic features of absorption machines. This enables the investigation of chosen application fields for their suitability for the installation of absorption chillers.

3.1 District heat

District heating means the central production of heat and the consequent distribution to consumers. In general, district heating systems consist of heat generation plants, facilities that keep pressure and volume in the required ranges, water-treatment facilities as well as district heating transport- and distribution grids. In order to transfer the heat to the consumer, district heating transmission stations are installed. By the help of central heat stations heat production and consumption are levelled. Water or water vapor thereby serve as transportation medium. Figure 3.1 shows a simplified scheme of a district heating grid from production of heat to the subsequent transmission to the consumer.[23]

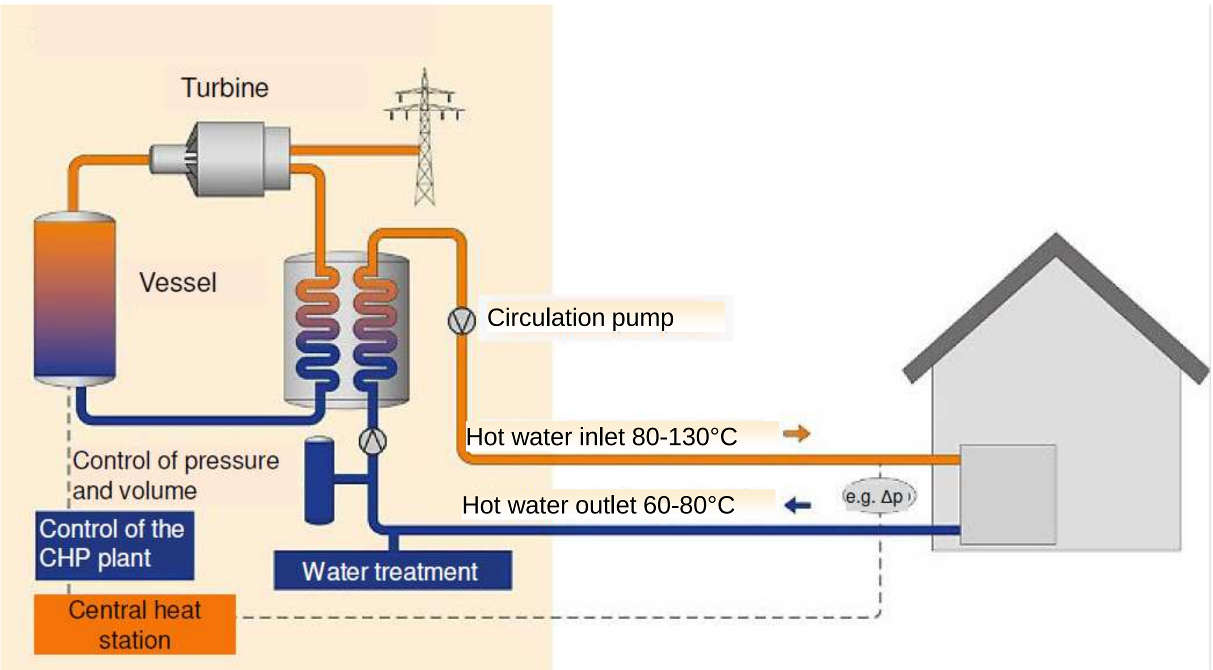

Figure 3.1: Simplified working scheme of a district heating system (translated from [23])

In Germany, the main part of district heat is produced in combined heat and power plants and the share amounts to more than 80%. In certain cities, such as Stuttgart, even 90% of district heat is produced via cogeneration. As for the fuel, district heating systems are flexible. Coal, natural gas, fuel oil, waste as well as biomass fired plants are used. Typical inlet temperatures of the district heating grids of EnBW are in a range of 105°C in winter months, when the demand is high and about 80°C in summer time. Outlet temperatures of the district heating grid should be as low as possible and are usually in a range of 60°C to 80°C.[24]
Figure 3.2, shows a qualitative district heat sales curve over the course of an entire year. Thereby the lack of heat demand in the warmer summer months can be well observed.

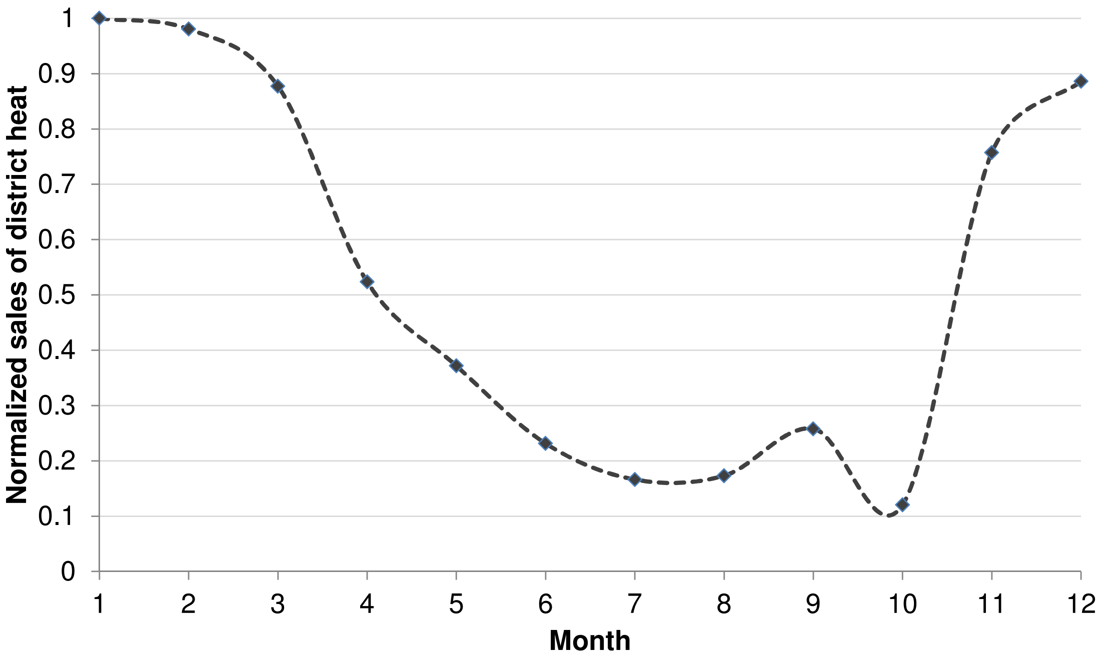

Figure 3.2: Exemplary annual sales curve of district heat

3.2 Optimal boundary conditions for absorption systems

Conditions that are advantageous or disadvantageous for absorption chillers are derived from the characteristic features of absorption machines at first. The following beneficial boundary conditions can be named:

- Constant base load in the respective cooling interval.
  ⇒ This means that the load profile should not display many sudden load changes and in general exhibit a preferably high amount of full load hours.
- Preferably high cold water temperatures T_0.
- Preferably low heat rejection temperatures T_C.
- Preferably high heat source temperatures T_{gen}.

However, the listed boundary conditions need to be treated with caution as they represent conditions for the absorption machine itself and not necessarily for the overall system. For example a constant base load over the course of the whole year leads also to an increased demand for heat in the winter months in which the district heating systems are generally already sufficiently used. Low heat rejection temperatures usually require generous dimensioning of heat rejection systems leading to both higher investment as well as operation costs. The heat source temperatures are determined by the district heating grid. Especially in summer - when the biggest cold demand is observed - the temperature level is at the lowest level. An increase in temperature in order to optimize the operational behaviour of absorption chillers is expensive and not practicable.

3.3 Evaluation of potential sectors

Absorption chillers can only compete with compression chillers when used in the right application, underlying a reasonable load profile which supports the benefits of the system. In order to identify sensible application fields, it is thereby important to know the requirements and characteristics of typical load profiles of different fields, such as food industry, chemical industry, air-conditioning of office/residential buildings, etc.. As figure 3.3 shows, e.g. in office buildings, a rather high increase of the cold demand in Germany is expected, which would require new installments of cold capacity. In order to determine the suitability of absorption chillers to take over parts of this cold provision as well as cold provision in other sectors, the requirements of these different sectors on the cold respectively typical load profiles of different sectors must be analyzed. As a matter of fact it is in general not feasible to pronounce the suitability of a whole branche for the equipment with absorption chillers. An assessment must always be performed separately under consideration of the specific boundary conditions of each individual application and load profile. As it has already been outlined in chapter **??**, basically solely lithium-bromide water machines exhibit attributes that are reasonable to be driven by district heat. For this reason, application fields in which cold temperatures lower than about 6°C are required, can be sort of excluded from the beginning. The investigation of interesting fields is conducted within the industrial sector as well as the sector of air-conditioning in buildings.

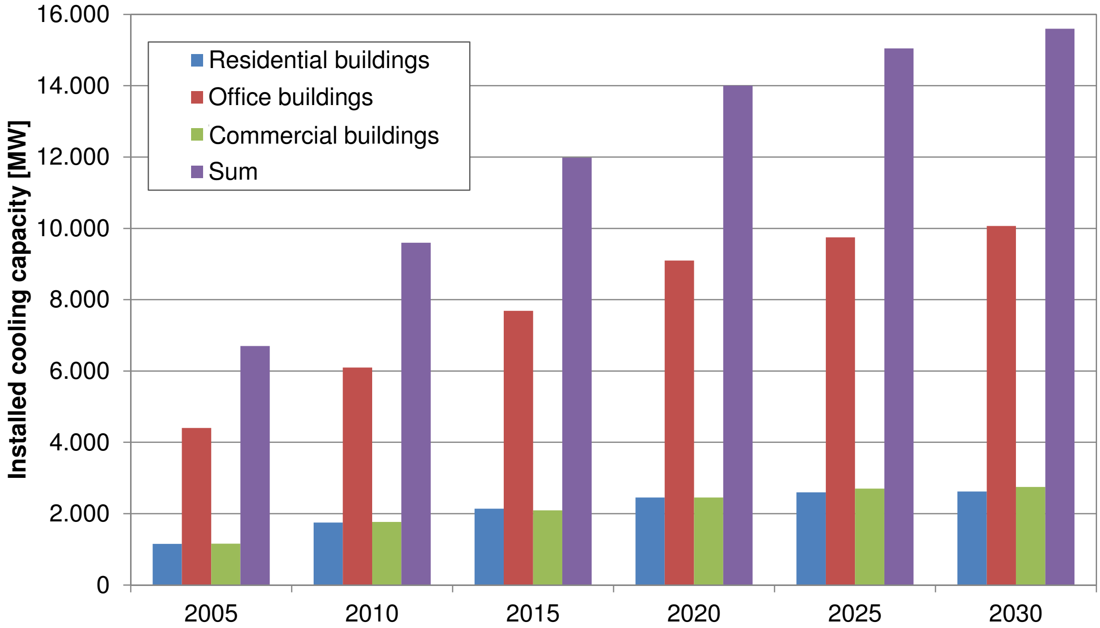

Figure 3.3: Possible development of the installed cold capacity in certain sectors in Germany (values from [25])

3.3.1 Industrial sector

Within the industrial sector a lot of branches can be excluded from the start, due to the required cold temperatures and the technical limit of the technology. This includes food production industry, chemical industry, construction- and building- materials industry as well as electrical industry [25]. Within industrial applications remain the fields automotive, mechanical engineering, process cooling, paper and cellulose industry, pharmaceutical industry and plastic and rubber processing that often require cold of about 6°C [25]. For this reason these possible branches in which absorption technology could be applied.

3.3.2 Air-conditioning of buildings

Within the field air-conditioning of buildings, the temperatures, that are usually required, fit into the working range of lithium-bromide absorption machines. The sector can be divided into the groups residential buildings and commercial-, office-, industrial- buildings as well as datacenters.[25]

Residential buildings

With respect to the permanence of which a certain cold demand is requested, residential buildings represent the most inconstant and unpredictable group within the sector air-conditioning of buildings. Furthermore, residential buildings usually have small cooling demands for which the higher investment costs of absorption machines preponderate and hardly can be compensated by the expected lower operational costs in comparison to compression systems. The cold demand in residential buildings is generally strongly dependent on the ambient conditions. Therefore the biggest cold demand would basically be reached in the daytime in summer. Residential buildings are however often empty/little occupied during daytime. All in all these reasons lead to the fact, that residential houses do not represent a promising target group for equipping the buildings with absorption machines.

Commercial-, office-, industrial- buildings and datacenters

In comparison to residential buildings, all of the above groups imply an improved predictability of the cold demand. Furthermore, the size of the buildings is often big enough to provide a sufficiently large cooling demand to promise a cost-efficient operation of absorption machines. The different groups of cold consumers, that are listed here, often cannot be strictly separated but occur in combination. For example, big office buildings often have server rooms in addition to the actual office space. The cold demand of pure office space, industrial- and commercial buildings in general can be characterized by a strong dependence on the outside temperature and thus have a high demand in summer and a rather low/non-existent cold demand in winter. Data centers and server rooms usually exhibit a rather permanent cold demand over the course of the year, that is less dependent on ambient conditions.
To support these assumptions, figure 3.4 shows four normalized annual load duration curves of two existing buildings. The curves were compiled by the evaluation of cold customers of EnBW and FUG Ulm [1]. These customers are currently served with cold, that is generated in compression chillers. For the analysis of the basic cooling demand the way of cold production is however thoroughly irrelevant.

The corresponding buildings for which the cold demand is shown are office buildings, each however having in addition server rooms and (in the case of the building represented by the red line) some pro-

[1] "Fernwärme Ulm" is the local district heating company of Ulm

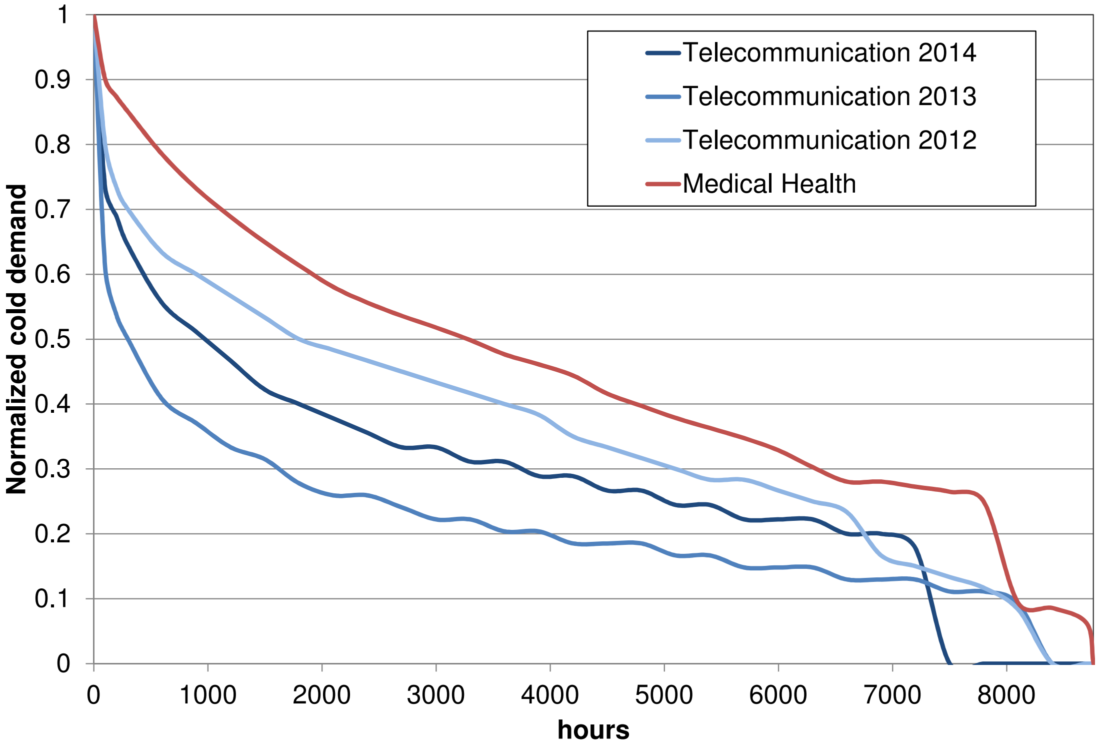

Figure 3.4: Exemplary annual load curves of chosen office buildings (normalized to their respective maximum cold demand)

duction facilities.
The red line represents the cold demand of the headquarter of a company working in the development and production of medical health equipment. The blue lines represent the cold demand for the years 2012, 2013 and 2014 of an establishment of a telecommunication company. For the year 2014, solely data until the midth of November were available and therefore the cold demand is temporally limited. A precise explanation of the remarkable differences in the cold demand between the different recorded years cannot be given. The cold demand is deduced by the data of the metering systems that have been provided by the respective utility companies. Possible reasons could be e.g. reorganizations within the company, differences in the ambient conditions, insufficient metering in the respective years as well as refurbishment of the equipment.

It is noticeable, that both buildings display a certain cooling demand over the course of the complete year - also during the nights. This basic demand is most likely due to the mentioned server performance and/or due to production heat that needs to be purged. Furthermore it is remarkable, that the rated refrigeration capacity, for which the systems are designed, are requested solely for a very distinctly proportion of the year.
Figure 3.5 shows the cold demand of the telecommunication building over the course of 4 chosen days that are distributed uniformly over the four seasons of the year 2012. For the two days in autumn/winter a very constant cold demand can be observed. It is very likely, that this is the basic cold demand of the server/data processing rooms. The other two days however show a significant variance in the cold demand during the day. This behavior is particularly pronounced on the day in midsummer (01.07.2012), where almost rated capacity is reached. The cold capacity that exceeds the basic demand of the data rooms is most likely associated to the ambient temperature depending share of the office rooms.

In summary it can be stated, that both, the industrial sector as well as the sector air-conditioning in buildings exhibit areas for which a basic suitability of LiBr-water absorption chillers is given. Nevertheless, certain subgroups can be excluded a priori for the reason of required cold water temperatures that fall short of limits of the technology - or due to inconstant and insufficiently large cold demands. An example is the food industry, which is a sector requiring high amounts of cold but at a temperature level below the application area of LiBr-water single stage absorption chillers.

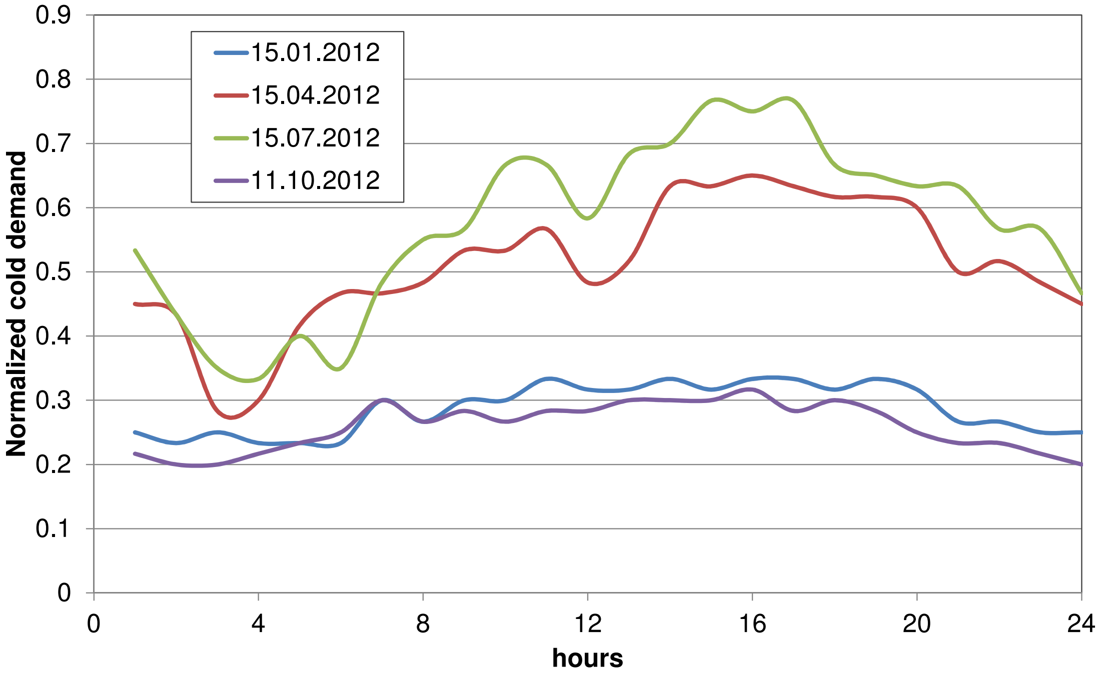

Figure 3.5: Cooling demands over the course of the day (Telecommunication - 2012)

4 Manufacturer review

This chapter aims to give an overview over absorption chillers that are currently available on the German market. Many important manufacturers and their affiliate distribution companies are briefly introduced as well as their product line. Special focus is on water driven, single-stage machines that are suitable for the use of district heat as driving energy. For this reason, technical specifications of absorption chillers driven by different heat sources will not be subject of this chapter. As it is of tremendous importance that the machines are retailed in Germany, only these machines are introduced. Most of bigger absorption chiller manufacturers are international companies, do not retail their products in Germany by themselves but use sales partners which are usually specialists in HVAC and support planning, consulting and communication. Manufacturers or their German sales partners have been contacted in order to get informations about the prices of the machines and to receive comparable performance data for uniform underlying assumptions.

The basic working principle and cycle configurations of the introduced machines are similar for most manufacturers. Nowadays all machines are equipped with an additional heat exchanger between the dilute and the concentrated solution. Figure 4.1 shows the basic underlying working scheme and the nomenclature of temperatures, which is used in the following. As a matter of fact, the machines of the individual manufacturers still have constructional differences. These are however not adressed in particular, as it concerns confidential matters of the companies on the one hand and besides is not essential for the evaluation of the performance of the different machines.
Table 4.1 gives an overview over the different manufacturers, their retailers as well as the performance range of their water-driven absorption chillers.

Table 4.1: Summary of manufacturers and retailers of LiBr-water absorption chillers, available in Germany (own compilation without claiming completeness, data from [26][27][28][29][30][31][32][33])

Manufacturer	**Performance range**	**Fluid combination**	**Retailer in Germany**
Yazaki	17.5 - 75 kW	LiBr/water	Gasklima, Johnson-Controls
World Energy	105 - 4571 kW	LiBr/water	Gasklima
York	280 - 3150 kW	LiBr/water	Johnson-Controls
EAW	15 - 200 kW	LiBr/water	-
Shuangliang	350 - 4650 kW	LiBr/water	Ruetgers
Zephyrus (Shinsung Engineering)	53 - 3511 kW	LiBr/water	Benndorf-Hildebrand GmbH
Thermax	60 - 3943 kW	LiBr/water	Trane
Carrier	264 - 1846 kW	LiBr/water	-

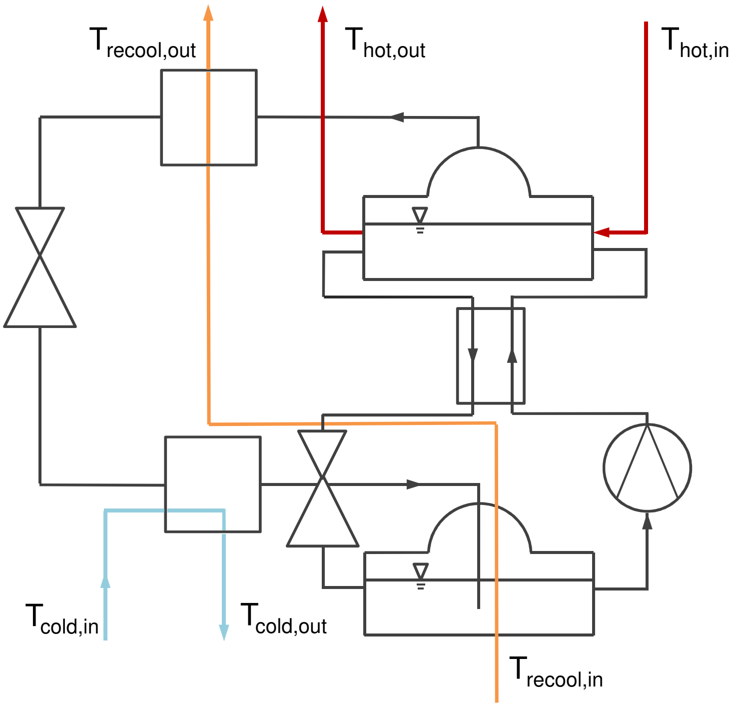

Figure 4.1: -
Typical, basic working scheme of the presented absorption chillers, underlying declaration of temperatures

4.1 Manufacturers

In the following, the companies are briefly introduced and an overview over the technical specifications of the machines is given. A more detailed overview over the different machines and their technical specifications is given in form of tables in the appendix.

4.1.1 Yazaki

Yazaki is a Japanese, private owned and worldwide acting company divided into the sectors automotive, environmental systems and new business fields. Within the energy/environmental sector, Yazaki has established as one of the biggest players in the market of absorption chillers. The products range from small to medium-sized absorption chillers that are driven by hot water or steam, or directly fired by gas. Today more than 100.000 Yazaki units are in operation worldwide from which more than 2000 are installed in Europe. In Germany, Yazaki's absorption chillers are retailed by Gasklima and Johnson-Controls. [26]

Water driven absorption chiller

The product line in the case of water driven absorption chillers consists currently of two types of machines (two series) each being available at five different performance classes, ranging from 17,5 kW to 175 kW of rated power and using lithium bromide as absorbency and water as refrigerant.
In Germany Yazaki machines are retailed by Gasklima which exclusively sells the WFC-SC series. This series is exclusively designed for cooling. The second series (WFC-SH) is very similar regarding its working scheme but has installed an additional changeover valve, making it applicable to both heating and cooling operation. The WFC series operates in a single cycle and is listed with a coefficient of

performance of 0.7 at rated power for all sizes. The machines are designed for rated heat source temperatures of 88°C and have a relatively small heat spreading of 5K, meaning that the hot water leaves the generator at about 83°C at nominal conditions. The acceptable hot water inlet temperatures range from 95°C to 75°C. The standard cooling cycle temperatures vary between the different models and lie between 27°C/32°C and 30°C/36°C.[34][35]

4.1.2 World Energy

World energy is a Korean developer and manufacturer of absorption cooling heating technologies, founded in 2004. Via qualified and experienced national distributors such as Gasklima in Germany, they provide local expertise and support. The product range regarding absorption chillers is very wide so that exhaust gas-, water-, steam- and direct driven heat pumps are offered as well as customized absorption chillers and heat pumps.[28]

Water driven absorption chillers

In the case of water fired absorption chillers, World Energy offers three different types of LiBr-water machines which in turn are available in different performance classes.
The 2AA - low temperature waste heat driven cooling machines are designed for low hot water temperatures T_{in}/T_{out} of 70°C/60°C. At designed working point, the inlet/outlet temperature of the chilled water is 13°C/8°C and the cooling water 31°C/36°C. According to World Energy, the COP of the 2AA series reaches values up to 0,45. The cooling performance of the machines ranges from 264kW to 4571kW.
The Single Effect Hot Water Driven Absorption Machine (HWAR-L) series ranges from 105kW to 4571kW. Chilled water temperatures and cooling water temperatures at design point are very similar to those of the 2AA and amount to 13°C/8°C and 31.5°C/36°C. The rated hot water temperatures are however higher and amount to 95°C/80°C at laid-out conditions. The COP is then about 0.72.
The third chiller series, the 2AB - high hot water delta-T driven cooling, was - according to the manufacturer - developed for district heating companies to maximize the use of hot water thermal energy by realizing a high hot water temperature difference between inlet and outlet of the chiller. Underlying is a single-effect, double lift cycle configuration. This allows smaller water flow rates and therefore smaller pipeline diameters and lower pump power can be realized. The 2AB chiller is standardly available within the power range of 264kW - 4571kW, reaching COPs up to 0.70. The standard hot water inlet/outlet temperatures are 95°C/55°C, the cooling water temperatures 31°C/36.5°C and the chilled water temperatures 12°C/7°C. [36]

4.1.3 York

York is a company, specialized in the manufacturing of refrigeration and air conditioning systems, which was found in 1874 in York, Pennsylvania. In 2006 it became a brand of Johnson Controls. York has three different types of chillers whereof two of them are two-stage systems, one of it directly fired and one vapor driven. The third one is a water driven machine.

Water driven absorption chillers

The water driven absorption series, YIA is available in 21 different performance classes, ranging from 280kW to 3150kW. It can be driven by vapor or water. The machines are laid-out for cold water temperatures at inlet/outlet of about 12.2°C/6.7°C, cooling water temperatures of about 29.4°C/38.5°C and heat source temperatures at the inlet of about 95°C. The rated COP ranges from 0.61 to 0.71 between the different performance classes.[29]

4.1.4 Carrier

Carrier is a subsidiary of the United Technologies Corporation, belonging to the business unit Building & Industrial Systems and being active in the field of HVAC.[27]

Water driven absorption chillers

The prodcut porfolio currently consists of one hot water driven absorption chiller series, being available in 18 different performance classes, ranging from 80 to 1846 kW. Basically the machines are conceived for heat source temperatures of 95°C and a heat spreading of 9°C making the hot water leave at 86°C. The machines are intented to provide cold of about 6.7°C. The temperatures for the heat rejection cycle are given by 29.4°C/38.4°C.

4.1.5 EAW - Energieanlagenbau

EAW - Energieanlagenbau is a German subsidiary of the WEGRA Anlagenbau GmbH situated in Westenfeld, Thüringen, producing energy systems for cogeneration, heat- and cold production.[32]

Water driven absorption chillers

Currently the product portfolio of EAW consists of one LiBr-water absorption chiller series being available in six different sizes. EAW coveres solely smaller cold performances in between 50kW and 200kW. The rated COP of all machines is in the same order of magnitude and is indicated between 0.71 and 0.75. The machines are designed for heat source inlet temperatures of about 90°C, with a temperature spread of 10°C for small machines and 15°C for bigger machines, meaning the hot water leaves the generator at 80°C respectively 75°C. The machines are designed to provide cold water with outlet temperatures of 11°C for the smallest machines (Wegracal SE 15 and SE30) and 9°C for the other four machines. All machines are equipped with hermetically sealed canned motor pumps and use plate heat exchangers.
Since the first machine has been installed in Rieth in 2001 EAW has sold several machines in Germany. Whereas the machines have been rather small in the beginning years, in the recent years merely the two bigger models have been installed. EAW often sells absorption chillers in combination with its CHP power plants. [32]

4.1.6 Shuangliang Eco-Energy

Shuangliang Eco-Energy Systems is a major subsidiary of the Chinese Shuangliang Group, providing solutions in energy efficiency, fresh water management and producing components such as high efficiency heat exchangers and absorption chillers, driven by various different heat sources. Today Shuangliang offers different types of absorption chillers such as steam operated, direct fired, flue gas operated and hot water operated absorption chillers, all using lithium bromide as working fluid.[37] In Germany the machines of shuangliang are retailed by the specialist in HVAC systems and industrial cooling - Ruetgers.[31]

Water driven absorption chillers

In the standard product line, Shuangliang has both a single-stage and a two-stage machine. As the two stage machine requires heat source temperatures at about 130°C in order to be operated efficiently it is not suited for district heat operation. The single stage hot water driven absorption chiller exists in 13 different performance classes, ranging from 350kW to 4650kW. The stated COP for all machines is 0.81 at design conditions, even so it has to be considered that this value is reached for cold water temperatures of about 15°C/10°C, cooling water temperatures 32°C/38°C and hot water temperatures 95°C/85°C during nominal operation. [37][31]

4.1.7 Zephyrus (Shinsung Engineering)

Shinsung is a Korean producer and developer of HVAC systems and components. Shinsung divides its products and services into four main areas, the comprehensive HVAC business to which the absorption sector belongs, industrial HVAC business, New Renewable / Environmental business and marine HVAC business. The HVAC products nowadays are produced by Shinsung-Engineering's brand "Zephyrus". In Germany the absorption machines are retailed by the Benndorf-Hildebrand GmBH.[30]

Water driven absorption chillers

In the following there is only a presentation of the water fired absorption chillers which are distributed in Germany by Brenndorf-Hildebrand. The product line consists of two different chiller types, the SAB-HW hot water driven absorption chiller and the SAB-LW Single Effect Double Lift Hot Water Driven Absorption Chiller, both operated by the fluid pair lithium bromide / water. The SAB-HW chiller is available in 25 performance classes. The COP can be calculated by the data given by the manufacturer to 0.71 at standard conditions and is the same for all performance classes. The standard chilled water temperatures at inlet/outlet are 13°C/8°C, the hot water temperatures 95°C/80°C and the cooling water temperatures amount to 31°C/36°C, also constant for all classes. [30][38]

4.1.8 Thermax

Thermax Group is an international company providing a wide range of engineering solutions within the energy and environment sectors. Thermax products range from cogeneration plants, heating equipment, air pollution control systems over waste heat recovery units to absorption chillers. The Thermax Europe Ltd, which is the European branch is mainly active in heating and cooling solutions.
With respect to absorption chillers, Thermax offers different machines, being operated either by hot water, vapor, gas/oil fired ones as well as flue gas driven machines. [33]

Water driven absorption chillers

Thermax offers three different hot water driven apsorption series, named "Cogenie", "Extended Cogenie" and "LT-T (WIN)". The machines covering the smaller cold performances are the Cogenie and Extended Cogenie which are available for performances from 60kW to 620kW. The LT-T(WIN) series covers the bigger performances and is available from 844kW until 3943kW. Constructionwise this series exhibits a singularity as the design can be described like two individual cooling machines being located in a single housing. All heat exchangers are splitted, the absorbers are working in parallel configuration and the lithium bromide flux are also lead parallel. The generators are however connected in row and enable therefore big temperature spreadings between inlet and outlet of the hot water. Depending on cold water temperatures and cooling water temperatures, hot water outlet temperatures down to 60°C can be realized in steady operation in the LT-T (WIN), whereas the Cogenie series just enable hot water outlet temperatures until 65°C. Hot water inlet temperatures are possible between 75°C and 120°C for all types. For rated operation parameters, meaning temperatures at inlet/outlet of hot water are 90°C/80°C, chilled water temperatures 12°C/7°C and cooling water temperatures of about 35°C/29°C, the COP of the machines varies between 0.68 and 0.75. [33]

4.2 Manufacturer inquiry

So far, the different manufacturers and retailers of water-fired absorption chillers as well as their product range have been introduced in order to get an overview over the market offer in Germany.
It is however very difficult to compare the individual machines among each other by simply comparing the data that is published by the manufacturers themselves. This is because often different boundary conditions like e.g. different temperatures are assumed. As it has been outlined in chapter **??**, e.g. the COP of absorption chillers is strongly dependent on the conditions and temperature levels of hot water, cold water and cooling water.
In order to achieve such a comparability, manufacturers were contacted to get informations and offers for certain specified operation conditions. These conditions were assembled to be quite different from each other to cover a possibly vast application area. Detailed information of the requested cases are shown in table 4.2. These include conditions that are more suitable for decentral CHP plants (higher heat source temperatures ≈ 95°C) and conditions that are convenient for district heat operation (lower heat source temperatures ≈ 80°C). The heat rejection temperatures have been chosen comparatively low, so that the heat sink system is specified for evaporative coolers. Cold water temperatures have been chosen first - rather low - to cold water temperatures that are typical for air-conditioning in older buildings (typical temperatures 6°C/12°C), and second to temperatures that are suitable for process cooling and/or air-conditioning in new buildings (9°C/15°C). For each case, three different performance sizes were requested to cover a possibly wide power range, from rather small machines of 100 kW up to big machines of 1 MW.

Table 4.2: Requested boundary conditions

	Application 1: Industrial cooling	**Application 2: Air-conditioning**
Performance	100kW/500kW/1000kW	100kW/500kW/1000kW
Heat source	CHP or district heat	CHP or district heat
Heat source temperatures:	CHP: 95°C/75°C District heat: 80°C/65°C	CHP: 95°C/75°C District heat: 80°C/65°C
Heat rejection temperatures	$T_{in}/T_{out} = 25°C/32°C$	$T_{in}/T_{out} = 25°C/32°C$
Cold water temperatures	$T_{in}/T_{out} = 12°C/18°C$	$T_{in}/T_{out} = 6°C/12°C$

The experts of the respective companies were able to select the appropriate machine for the given conditions respectively adapt the machines to those.

Results of the manufacturer inquiry

Table 4.3 summarizes the filtered data from companies of which offers have been received[1], for heat source temperatures implied by the heat source "‘district heat". The names of the companies are thereby

[1] For the rest of the manufacturers, no suitable informations were available to the required date

not explicitly given, but named Manufacturer 1-4 (in the following also called Man1-4), since an allocation of the data to a specific company would not bring an added value. In addition to operating parameters, investment costs were requested in order to be able to compare them with each other and with cost functions from literature. In figure 4.2 the specific investment costs of the machines, provided for air-conditioning applications are shown. One manufacturer does not provide machines up to 1000kW that is why solely indications for machines up to 200kW were provided.

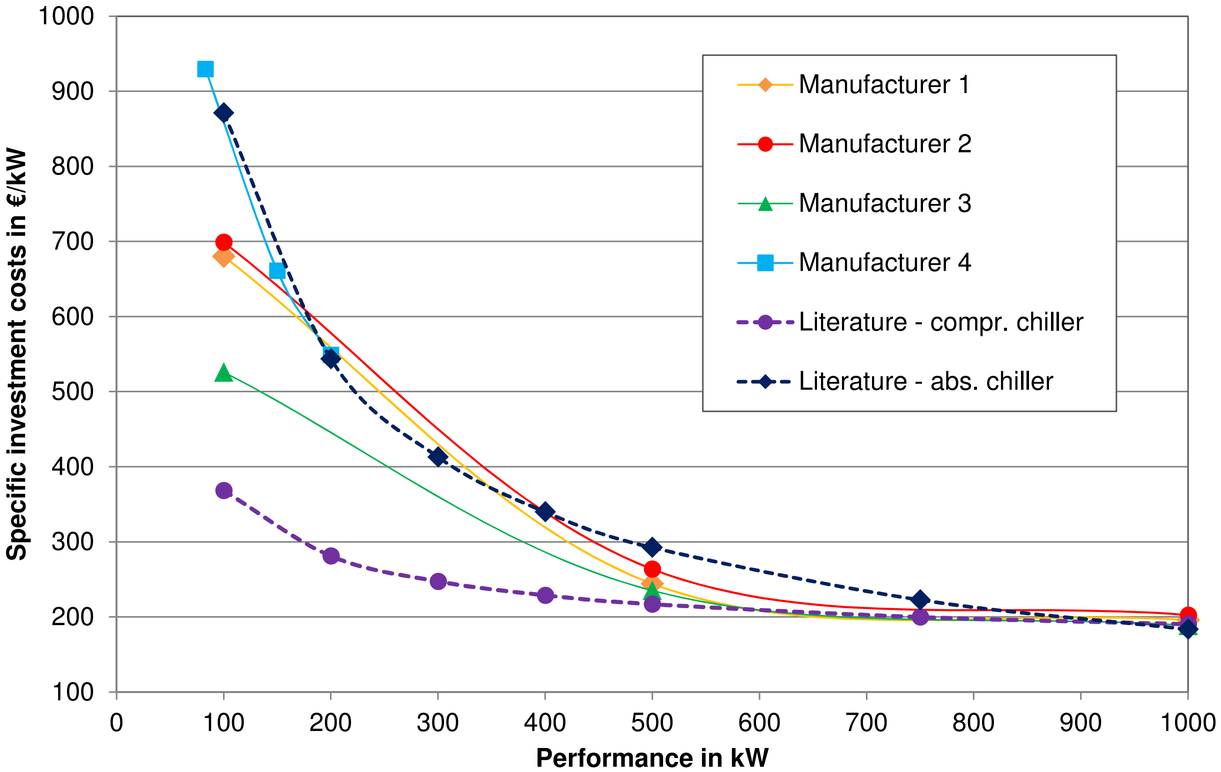

Figure 4.2: Specific investment costs of absorption chillers, (Application 2, district heat)

According to informations given by Man 1, their smaller machines are not capable of complying with the requested heat spreading at hot water side and consequently are not able to reach the heat spreadings of Man3 and Man2 machines. The heat spreading that is given by the Manufacturer 2 and 3 are well suited for district heating operation, because a hot water outlet temperature of 60°C-65°C is optimal. The inherently lower COPs of the machines of Manufacturer 3 in comparison to Manufacturer 1 can be explained by the lower hot water outlet temperature and therefore the higher heat utilization of their (Man3) machines.

Comparing the specific investment costs, remarkable differences between the different manufacturers exist (see figure 4.2). The 100kW machine of Man3 is with about 500€/kW considerably cheaper than the one of Man1 ($\approx$ 670€/kW), Man2 ($\approx$ 700€/kW), Man4 ($\approx$ 930€/kW) and the estimation given by the function from literature ($\approx$ 880€/kW). The cost function from literature for absorption chillers (cost function after [39], cf. chapter 6) is in accordance with the specific costs of the machines of Man4 for small performance classes. For bigger cold performances it is comparable to the other manufacturers, but rather exceeding. Compared to the specific investment costs of compression chillers (cost function after [39], cf. chapter 6), the investment costs of absorption chillers are principally higher. The smaller the performance of the respective systems, the bigger is this difference between the technologies. For machines $\geq$ 600kW, the difference is - if at all - negligible small.

In the appendix, the respective data are given for higher heat source temperatures ("CHP") (cf. table

Table 4.3: Overview over the data gathered in the inquiry (Heat source: District heat)

	Man1	Man2	Man3
100kW			
COP	0.69 0.75	0.74 0.76	0.0.67 0.73
Cold water temperatures	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C
Heat rejection temperatures	25°C/30.2°C 25°C/32°C	25°C/32°C 25°C/32°C	25°C/32°C 25°C/32°C
Hot water temperatures	80°C/72°C 80°C/68°C	80°C/65°C 80°C/60°C	80°C/66°C 80°C/63°C
El. power consumption	5.1kW 5.1kW	3.1kW 3.1kW	3.2kW 3.2kW
Spec. investment costs	680€/kW 526€/kW	699€/kW 673€/kW	526€/kW 520€/kW
500kW			
COP	0.74 0.75	0.74 0.76	0.69 0.73
Cold water temperatures	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C
Heat rejection temperatures	25°C/32°C 25°C/32°C	25°C/32°C 25°C/32°C	25°C/32°C 27°C/34°C
Hot water temperatures	80°C/70°C 80°C/70°C	80°C/65°C 80°C/60°C	80°C/65°C 80°C/64.4°C
El. power consumption	6.9kW 6.9kW	10kW 10kW	7.3kW 5.9kW
Spec. investment costs	244€/kW 213€/kW	264€/kW 264€/kW	235€/kW 191€/kW
1000kW			
COP	0.79 0.82	0.74 0.76	0.69 0.73
Cold water temperatures	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C
Heat rejection temperatures	25°C/32°C 25°C/32°C	25°C/32°C 25°C/32°C	25°C/32°C 27°C/34°C
Hot water temperatures	80°C/68°C 80°C/66°C	80°C/65°C 80°C/60°C	80°C/65°C 80°C/64.4°C
El. power consumption	11.2kW 6.9kW	16.4kW 11.3kW	7.3kW 7.3kW
Spec. investment costs	196€/kW 155€/kW	202€/kW 176€/kW	189€/kW 144€/kW

A.8). The investment costs for these machines are significantly lower as for the here presented machines, laid-out for district heat hot water temperatures. Remarkably is that for performance classes bigger than about 500kW, they even fall below the investment costs of compression chillers (according to the cost function given in literature [39]. Furthermore, these machines imply generally better efficiencies (higher COPs).
Figure 4.3 shows the partial load behavior that is indicated by Man1 and Man2 for their absorption machines as well as the curve from literature [11] (see also figure 2.9). It can be seen, that for all curves, the differences in COP at partial load operation is in a range of $\pm$ 10% of rated COP. Man1 specified two different curves for the partial load behaviour of their machines. One represents the standard machines, that are not equipped with and one that are equipped with an inverter. An optional equipment with an absorbent pump's inverter that controls the flow rate according to the given load is often possible in return of a certain upcharge [2]. Thereby savings in input energy and electric power consumption can be reached which can even lead - like in the presented case - to an increase of the COP, when operated in

[2]For Man1 this upcharge amounts to about 5000€per inverter

partial load. The standard partial load behaviour, without inverter leads however usually to a decrease of the COP with decreasing operation load like this is the case for the curve in literature.

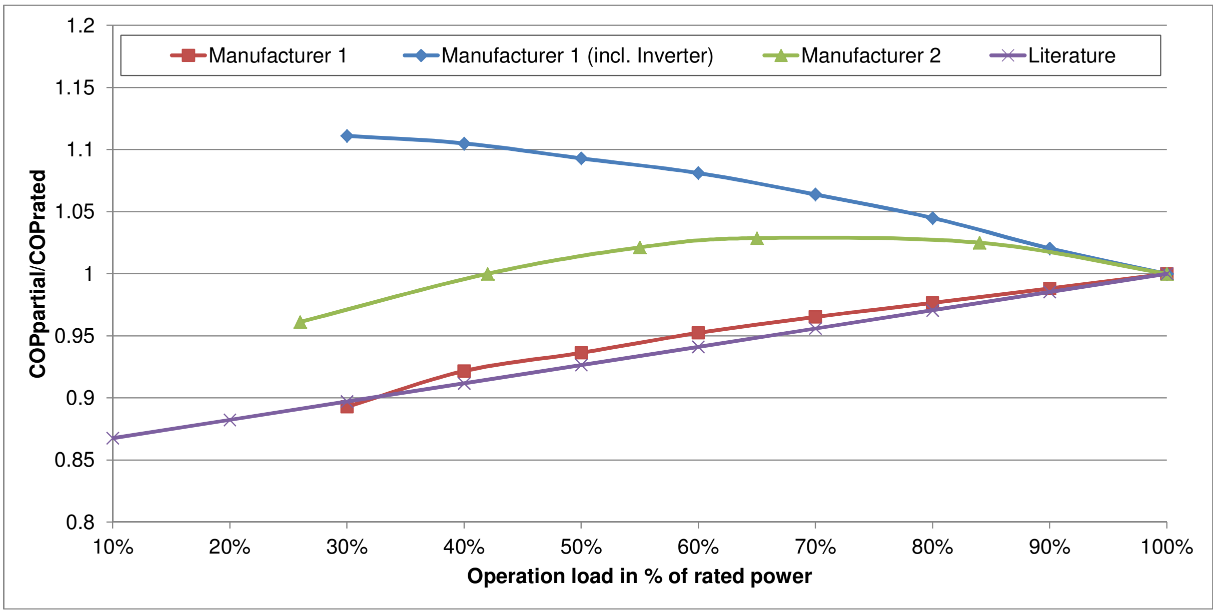

Figure 4.3: Partial load behaviour of different absorption chillers. Deduced by indications of manufacturers

An important piece of information for the installation of absorption chillers is the minimum required respectively the recommended size of the cold water buffer storage. Hereby the manufacturers made very different indications. Man4 recommends to dimension the storage by equation 2.20 for rated cold performance and a minimum bridging time of 15 to 20 minutes. Man1 stated that they do not use a fix calculation formula but recommend a storage capacity of 6 to 8 litre per kW installed cold capacity. The same applies for Man3 which state that for air-conditioning applications at least a buffer capacity of 3.25l/kW and for process cooling of 7.5l/kW is necessary. The indications of Man1 and Man3 lead thereby to significantly smaller buffer storages. If laid out at minimum indications they act more or less as hydraulic compensator but not as a real buffer.
Summarizing it can be stated, that the specific investment costs of absorption machines of different manufacturers exhibit remarkable differences for small cold performances, which are mitigated the bigger the machines get. Furthermore the specific investment costs of smaller absorption chillers exceed those of compression chillers by many times. For this reason, the influence of investment costs increases, the smaller the machine gets and consequently cannot be compensated by potentially lower ongoing and operating costs. This leads to a strong limitation of possible application fields of absorbers, making only large scale systems economically feasible (being to be proved in chapter 6). With regard to the operating parameters, slight differences in the COP of the machines can be observed which however cannot be approved in the literature due to a lack of real operating data (see however chapter 5).

5 Practical example

The previous chapters explained the operational behavior of absorption chillers based on theoretical knowledge from literature and indications given by manufacturers. In order to validate these informations on a real operating example, such an example is introduced and its behavior described and evaluated in the following. Presented is a contracting project of the municipal utility Stadtwerke Karlsruhe. After the complete refurbishment of an office building of a big insurance company in 2010/2011, Stadtwerke Karlsruhe got the mandate to renew an existing cold supply system. The chosen example is in particular suitable since it is a recently new installation with an up to date absorption machine.

5.1 Building and cold demand

The building that is to be climatized has a floor area of about 8500m^2 of which a major part is used as office space, but it also includes server rooms that ensure a constant cold demand of about 20-40kW over the course of the entire year. The contractually guaranteed cold supply amounts to 850kW at a guaranteed cold water temperature of 7°C. In the course of the refurbishment, the entire building was equipped with concrete core activation. This leads to a good insulation of the building, consequently mitigates peak loads and equalizes the overall annual cold demand. Figure 5.1 shows the cold demand of an entire year, from 10/2013 to 10/2014. Looking at this plot with a very rough resolution, it can be seen, that in the winter months the load basically consists of the roughly 20-40kW that are needed in order to cool the server rooms. During the rest of the year, the office space needs to be climatized as well and consequently the cold demand rises. In the summer months, the average cold demand is about 200-250kW and therefore still significantly lower than the contracted cold demand. Nevertheless several isolated load peaks occur that amount to 400-700kW.

Figure 5.2 shows the cold demand over the course of about 72h in summer (red line), beginning at the 15.07.2013. During the day and at maximum external temperatures of about 28°C - 30°C, the cold demand amounts to roughly 260kW. On the third day that is shown in this figure, the maximum temperature exceeds 30°C and consequently the cold demand rises, so that during this hottest time of the day, the cold demand amounts on average to about 330kW. Isolated peak loads occur but it can be seen, that these peak loads are usually merely recorded over a very short time period. Most likely they are associated with the temporary start-up of additional consumers.

In figure 5.3 the same curves are recorded for 4 days in spring, the red curve again representing the cold demand. It can be seen that in the night, there is only the need for refrigeration of the server rooms. During the day, the demand rises somewhat to values of about 100-200kW, having very short peaks up to 350kW.

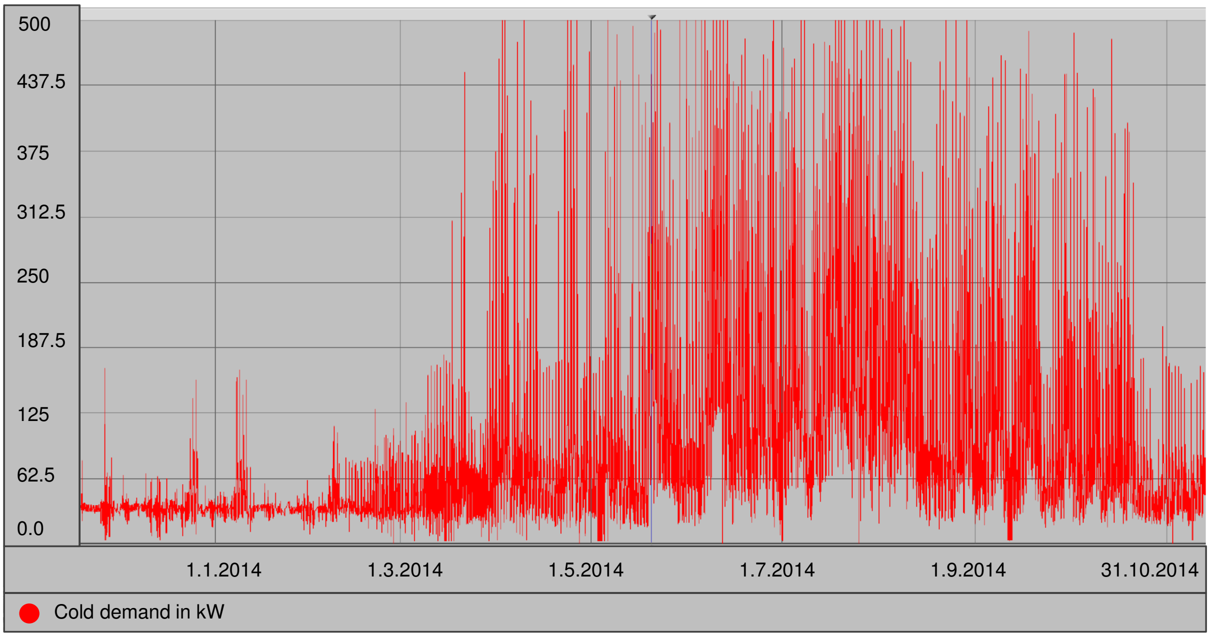

Figure 5.1: Cold demand over the course of a year

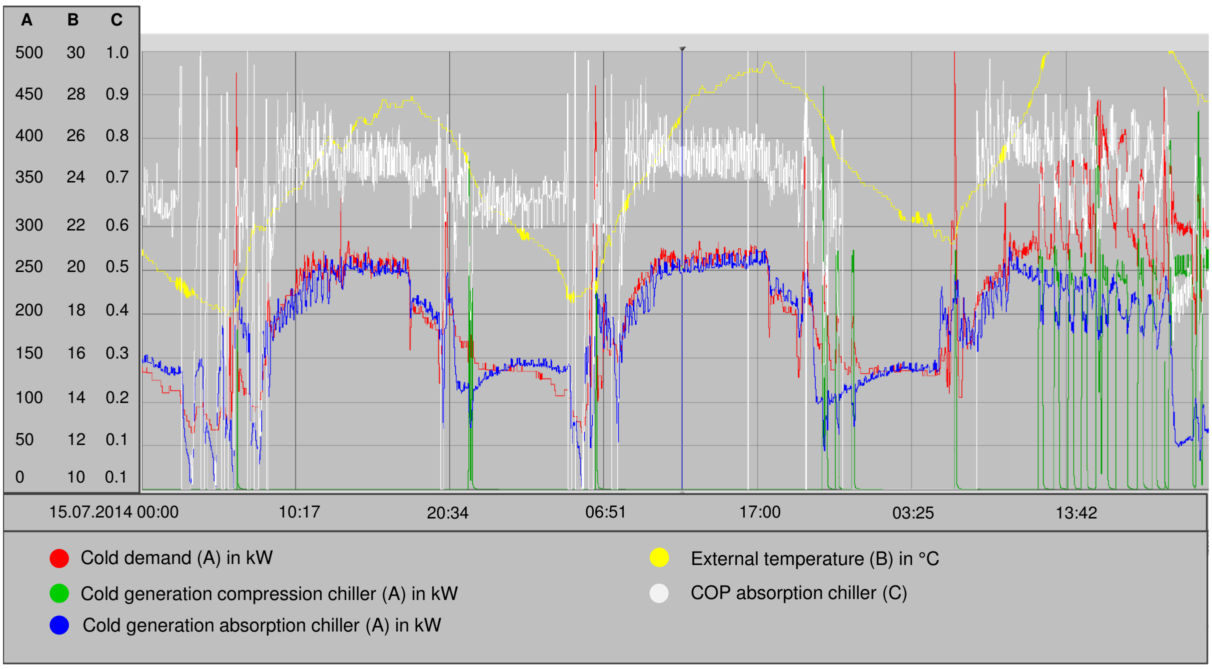

Figure 5.2: Operational informations over the course of 3 days (18.07.2014 - 20.07.2014)

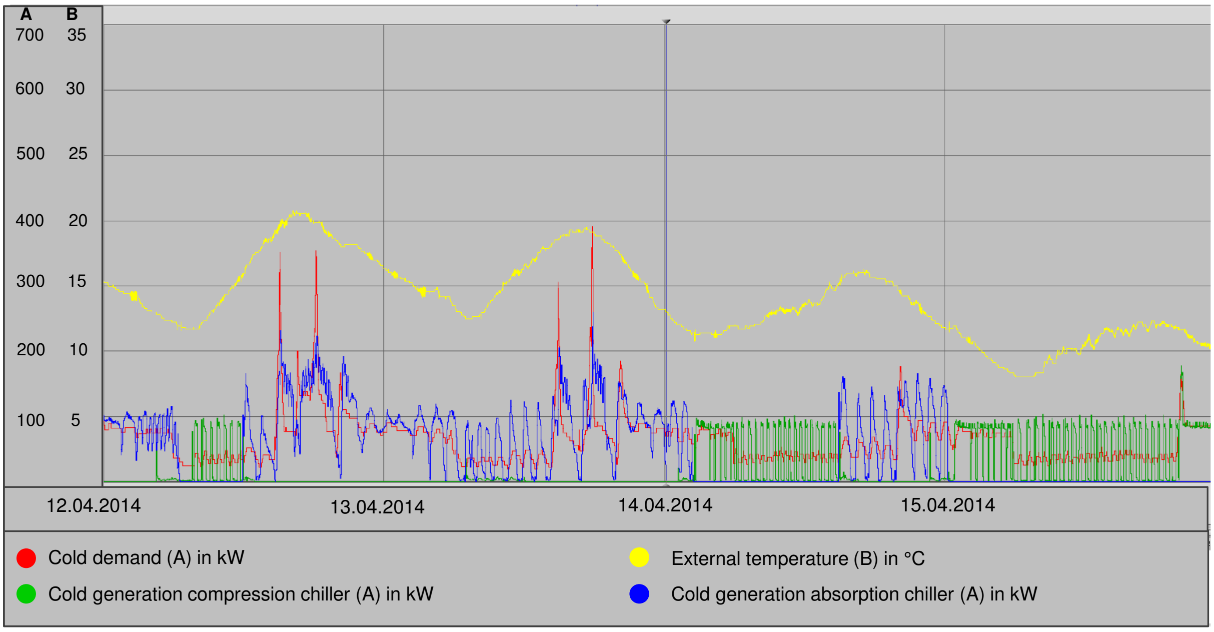

Figure 5.3: Operational informations over the course of 4 days (12.04.2014 - 15.04.2014)

5.2 Cold concept

Originally, the air conditioning supply in the building was comprised of two YORK LCHM 100 AL reprocating piston compression chillers - each having a cold capacity of 283kW. The machines are from 2004 and use the refrigerant R 407c. In the course of the re-equipment of the cold supplying system, these machines were taken over as supplementation to the new absorption machine. The absorption chiller that has been chosen is a Carrier TSA-16LJ-32P-LC unit, having a rated cooling capacity of 335kW for the specified boundary conditions. These laid-out conditions of the machine are shown in table 5.1.

Table 5.1: Laid-out conditions of the absorption chiller (Carrier TSA-16LJ-32P-LC)

Hot water inlet/outlet	82°C/63.5°C
Cooling water inlet/outlet	27°C/31°C
Chilled water inlet/outlet	12°C/7°C

For the heat rejection of the absorption chiller, it is made use of a hybrid cooling tower of the type KAVH by the company "Kühlturm Karlsruhe", having a capacity of 960kW. The heat rejection of the compression chiller is separately realized by the dry heat rejection system that has already been installed in the first place.
In order to compensate sudden load changes, a 8,6m^3 cold water stratified buffer store is integrated into the system. The shortage of space thereby limited the size of the cold storage which otherwise would have been bigger. A plate heat exchanger with a capacity of 800kW is furthermore installed and connected to the hybrid cooling tower and the cold storage (and consequently the cold consumer) enabling free cooling in the winter months.

Figure 5.4 shows the integration of the individual components into the overall system.
It can be seen, that the cold consumer is solely and directly connected to the buffer storage, meaning that all cold streams are flowing into the cold storage. The two compression chillers are connected parallel and can be regulated separately.

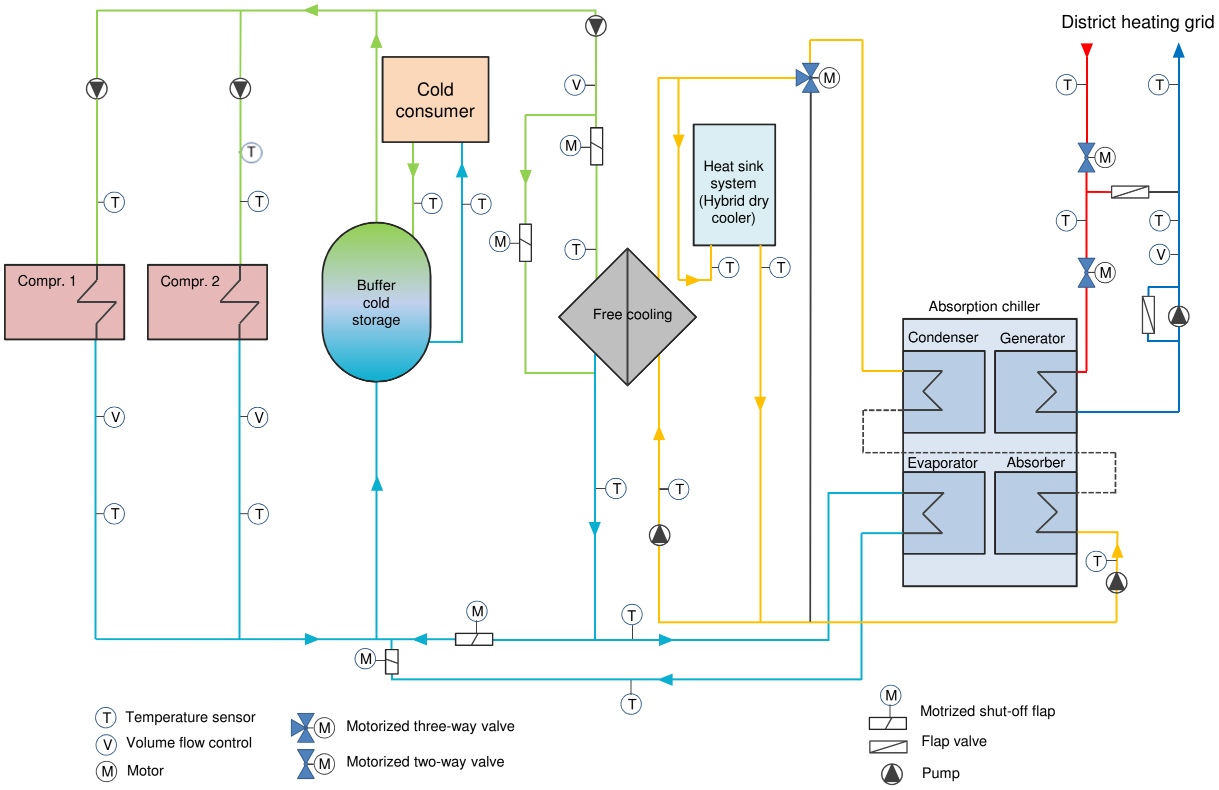

Figure 5.4: Simplified working scheme of the cold provision system

The evaporator of the absorption chiller is connected to the buffer storage system. The condenser and the absorber are directly connected to the heat rejection circuit. The absorber receives the recooling water at minimum temperature, which is then lead into the condenser, where it leaves at maximum temperature. Consequently it is lead through a three-way valve and is fed back into the absorber and/or into the cooling tower. The generator of the absorption machine is connected to the district heat. Because the hot water temperatures are already quite low during the summer months, a direct connection is needed in order to avoid a further decrease as a reason of the for heat transfer necessary temperature spreading in indirect systems. The maximum hot water inlet temperature of the absorption chiller is determined to 98°C by the manufacturer. For the reason that the temperature of the water that is provided by the district heating grid can exceed this value in the colder months, a security valve needs to be included in order to eliminate the risk of overheating. Furthermore, a mixing control circuit is integrated before the entry into the generator. By the help of this interconnection, hot water return flow can be mixed with hot water inlet flow and the temperature can be kept at the right (laid-out) level.

5.2.1 Control of the absorption chiller

The cold performance of the absorption chiller is regulated by the position of the hot water control valve. By the adjustment of this valve, the amount of heat that is supplied to the desorber is automatically adapted to the required cold demand. By the reduction of this supplied heat in partial load operation, the outlet temperature of the generator is reduced automatically. This thereby reduces the concentration of the solution at the outlet of the generator, leading consequently to a lower absorption ability in the absorber and at the same time reduces the cold performance of the absorption chiller. The volume flows of the cold and cooling water have to be constant due to specifications of the manufacturer.

5.2.2 Control of the overall system

The overall system has a parent, prioritized task, the delivering of the exact amount of cold at the concerted temperature that is required by the consumer. The contractually guaranteed temperature amounts in this project 7°C. This temperature is controlled in the buffer storage system in which the respective temperature sensors are integrated. Which cold generation device is used in order to produce the demanded cold is in general dependent on the external conditions. For low external air temperatures (≤5°C[1]), the cold supply is realized by the free cooling system. For external air temperatures higher than 5°C, but lower than 10°C, transition operation is active and the two compression chillers are accountable to provide the base load. If the capacity of the two machines is not sufficient, the absorption chiller is added as peak load machine [2]. The absorption chiller then serves as base load machine as soon as the external temperature exceeds 10°C, the compression machine respectively taking over the peak loads. When the base load supply is switched from compression chiller to absorption chiller, the compression chiller(s) is (are) still running in parallel as long as the absorption chiller provides cold water at 7°C. In order to prevent permanent switching between the different modes, the external temperature is calculated as a delayed average value and therefore the actual switching between the modes is delayed in time.

When the free cooling mode is activated, cold water and cooling water are directed - by positioning of the changeover valves - through the plate heat exchanger. The cold water then cools down by releasing heat to the cooling water.

The system is controlled in a way that in case of a disturbance at one of the cold generators, it is automatically switched to another one. As first option in order to replace the free cooling system, the compression chiller is determined. The compression chiller and the absorption chiller replace each other. In case that the cold water inlet temperature exceeds 11°C an emergency cooling is requested. This circumstance is however to be avoided as it involves high costs for Stadtwerke Karlsruhe.

5.2.3 Housing of the system

A critical, limiting parameter of the whole project was the range of space that was given in order to place the cooling machines and its periphery. The compression chiller was already allocated in a cellar room outside of the building, next to the parking lots of the building. Beneath this room that was already used to its maximum by the accommodation of the compression chiller and its heat rejection fan, there was made room for the placement of the absorption chiller, the district heat transfer station [3], the cooling water treatment facilities, the control cabinets as well as the cold storage. Available were four adjacent cellar rooms. By knowing the spatial conditions, the absorption chiller and the cold storage were dimensioned as large as possible. The biggest room with about 25m^2 houses the absorption chiller and its necessary peripheral equipment. The water treatment plant and the buffer storage are placed in another room having about 15m^2. The control cabinets and the district heat transfer station are placed separately in small rooms.

The hybrid cooling tower is integrated into the parking space as well.

[1] This value can be adjusted individually in the control settings.

[2] This scenario is exceptionally unrealistic, as the cold demand is very unlikely to be that high considering such low external temperatures

[3] The station is solely needed for the connection of the regular district heat supply. The connection of the absorption chiller to the district heating grid is a direct one in order to avoid a drop in temperature due to the temperature spreading that is necessary for the heat exchange.

Figure 5.5: Housing of the absorption chiller and its peripheral equipment

Figure 5.6: Housing of the cold water storage,the water treatment facilities (left) and the hybrid heat rejection system (right)

5.3 Operating characteristics and results

Figure 5.2 is well suited to illustrate the operational characteristic of the cold supply system. The yellow line represents the external temperature that varies in this case between 18°C and more than 30°C. It can be seen, that the course of the cold demand (red line) is qualitatively well adapted to the course of the external temperature. The blue line represents the cold performance of the absorption chiller while the green line represents the compression chiller. The white curve plots the COP of the absorption chiller. In case of the COP, the heavy fluctuations around a certain value can among others be explained by time-displaced measurements of cold and heat supply. As this value is calculated by those two values it can cause distortion of the value.

During the first two days that are depicted in this figure, it can be observed, that the absorption chiller can provide almost all of the required cold. Short-term differences between cold supply and demand can be compensated by the buffer. Merely bigger deviations need to be provided by the compression chiller, though they occur rather seldom. Noticeable is that on both the 16.07. and 17.07. at around 21.00 o'clock an extended falling below of the cold demand can be observed, that is too big in order to be compensated by the storage and therefore needs to be offset by the compression chiller. This can be explained by the setup of the hybrid cooling tower. Beginning at 6 o'clock pm, the frequency limiter intervenes and diminishes the frequency of the fan from 38Hz to 30Hz as a result of sound emission guidelines. At 9 o'clock pm the frequency is further lowered to 25Hz and the operational mode

is mandatory dry [4]. As in this case at 9 o'clock the temperature is about 24°C, the operation mode switches from wet to dry and consequently the heat rejection temperature increases for a certain period. This leads to a decrease of the cold performance of the absorber and thus to a bigger deviation between cold supply and cold demand. The compression chiller then needs to startup operation in order to compensate the difference. On the third day - the 18.07.2014 - the compression chiller has to take over much more of the cold supply. The higher external air temperatures lead to a higher cold demand and apparently the absorption chiller is not able to comply with fluctuations at such high cooling capacities.
Looking at the COPs, that are reached during these 3 summer days, one can observe a relatively good and constant behavior. At constant performances of about 250kW, the COP amounts to approximately 0.75. For constant loads at smaller performances the COP is lower but still in acceptable range at about 0.65. At very unsteady conditions and big changes in cooling capacity, the course of the COP exhibits vast deviations which can - as already previously mentioned - be explained by inaccuracies caused by the measurement method, as well as the bad dynamic behaviour of absorption machines.
Figure 5.3 shows the cold supply of 4 days in spring. On closer inspection, one can observe that during this time it is switched between compression chiller and absorption chiller as base load machine. Responsible for the takeover of the base load by the compression chiller is the outside temperature. In this case, the temperature interval for transition operation (meaning the compression chiller takes over the base load) has been extended up to a temperature of around 12.5°C. When the external temperature falls below this value, the compression chiller takes over with a delay that is due to the delayed average value of the temperature. When the 12.5°C is exceeded for a certain period, the absorption chiller respectively takes over again. The peak loads that occur during absorption chiller operation are small enough to be compensated by the buffer storage without a need to switch by the compression machine. The curve of the COP of the absorption machine is not available as a consequence of the missing metering equipment at this time.
When the resolution of the recording is increased and solely 24h are displayed like in figure 5.7 the function of the cold buffer can be well depicted. During operation, there is a permanent falling below and overrunning of the cooling demand by the cold generation. This means, that the cold buffer is permanently loaded (violet area) respectively unloaded (yellow area).

5.4 Evaluation and optimization possibilities

Evaluating the overall performance of the absorption chiller, satisfactory results can be observed. The COP that is reached during more or less steady operation conditions is in the range of what has been stated by the manufacturer. The absorption chiller dominates the cold production over the course of the year. Figure 5.8 shows the monthly cold production of the different cold generation components from June 2012 to October 2014, as well as the corresponding values of the COP of the absorption chiller. It can be well observed, that the share of the provision by the absorption chiller has increased in the last year in comparison to the previous one. Between October 2012 and October 2013, about 51% of the cold have been produced by the absorption chiller, whilst the compression chiller produced 48.7% of the overall cold. Between October 2013 and October 2014, the share of the cold production of the absorption chiller could be increased to about 76%, whilst the compression chiller only took over about 13% and the free cooling system respectively roughly 11% of the overall cold production. In the winter months, the absorption chiller is generally barely used, which is most likely due to the fact, that the demand for heat is already high and consequently the district heating grid is not further stressed. Furthermore, a

[4]Usually the heat rejection system starts sprinkling when the external air temperature exceeds 19°C/20°C. For very humid air conditions, the wet operation starts even sooner

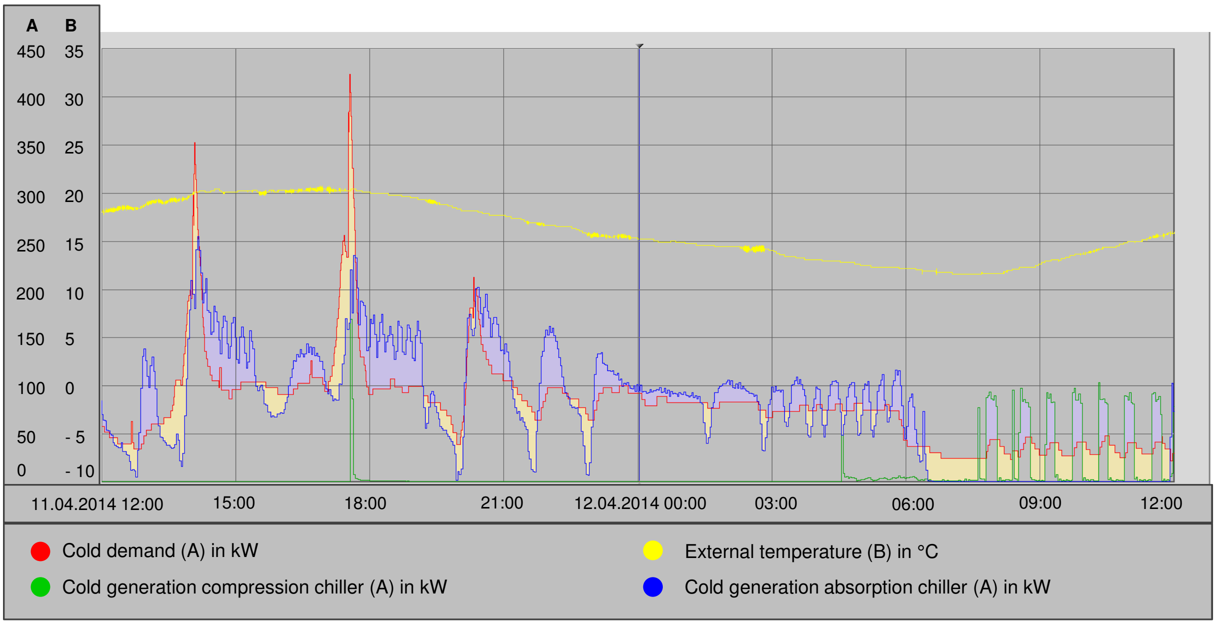

Figure 5.7: 24h resolution of cold demand and cold supply

general increase of the COP in the second of the considered periods, can be observed. From October 2012 to October 2013, the yearly average effective COP of the absorption chiller is about 0.56. In the consequent year, the value could be increased to about 0.62. It can be seen, that the COP reaches its highest values during the summer months and respective low values, in winter months. The increase of the COP and the load factor of the absorption machine can be traced back to slight adjustments regarding the control of the system, which have been made on the basis of operational experience of the system. A further increase of the utilization of the absorption chiller - and probably also the COP - would be possible, if a bigger buffer storage could be integrated.

When comparing the size of the storage system with the recommendations of the manufacturers one can see that the storage share for the absorption chiller which is 4.4 m^3 (after deduction of the size of the storage system, that is recommended by formula 2.20 for compression machines (4.2 m^3)) exceeds the recommended values of Man1 and Man3 (Man1 $\approx$ 2,4 m^3, Man3 $\approx 1{,}2$m^3) but falls clearly short of the values that are recommended by Man4 ($\approx$ 12 m^3). In the light of the above, the recommendations of the storage sizes given by Man1 and Man3 appear clearly undersized. If the buffer would be in such a range, the necessity of switching by the compression chiller would be given much more frequently. Consequently the efficiency of the system would suffer. The stuff in charge of the cold supply system at Stadtwerke Karlsruhe even expressed regret that it was not possible to install a considerably bigger buffer storage due to the shortage of space.
Further potential for improvement in the operation of the absorption chillers lies, according to Stadtwerke Karlsruhe, in the variation of cold water and cooling water flows. As it is the case for other manufacturers, Man3 wants to keep those volume flows constant. For this reason, during warranty no adjustments have been done, however as soon as this warranty period expires, it is planned to make adjustments here.
It is a declared target of Stadtwerke Karlsruhe to further increase the share of cold that is produced by the absorption chiller in the warmer seasons by additional optimization of the control of the system. In the winter months, the cold demand is then provided by free cooling operation.

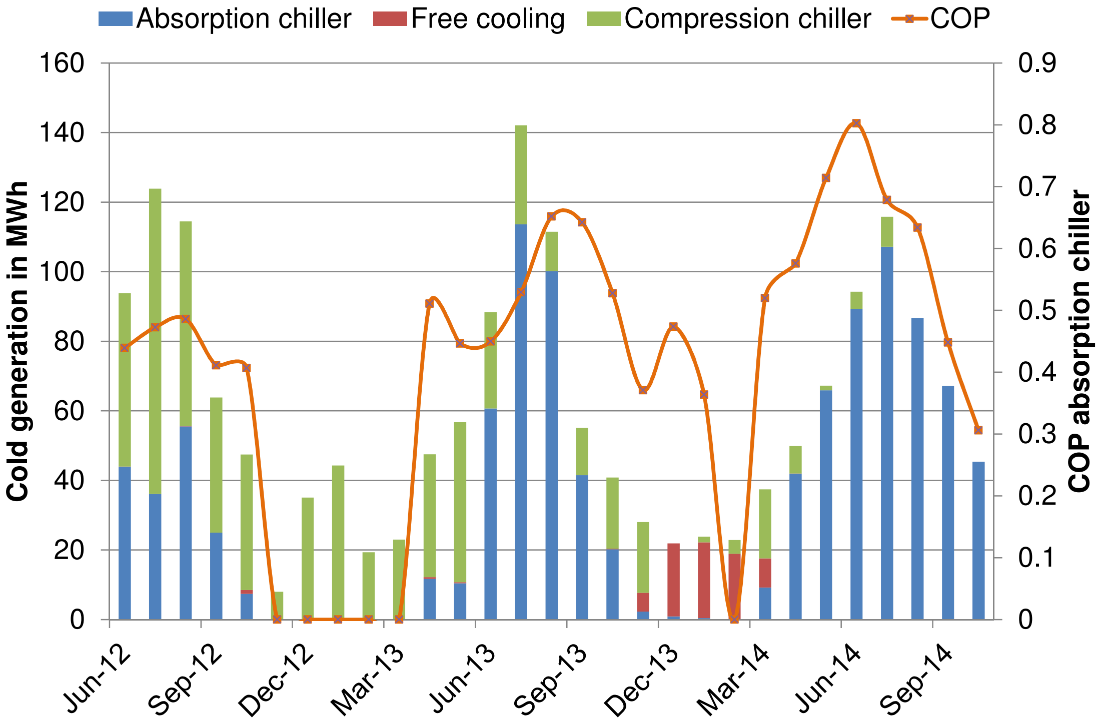

Figure 5.8: Way of monthly cold production and assignment to the different system components; Monthly average COP of the absorption chiller

5.5 Conclusion

The experiences that Stadtwerke Karlsruhe made - and which they could approve also in different projects - confirm the assertion that each project has to be assessed separately. Besides the example shows, that in addition to obviously decisive factors such as the load profile, there are also other factors that have an substantial influence on the success of the system. For example spatial conditions can inhibit or restrict the application of absorption machines and possibly optimal laid-out conditions cannot be implemented. Furthermore, adjustments regarding control settings and optimal operating conditions are time consuming tasks, that often happen over a period of months but rather years.
The example shows the tremendous importance of the size of the buffer storage on the load factor and the COP of the absorption chillers. The indications provided by Man1&3 appear clearly undersized and a dimensioning like proposed by Man2 desirable. However in particular the storage requires much space and consequently might often be limited.
The effective COP of the system shows with about 0.62 good agreement with indications given by the manufacturers and can probably even be increased in the consequent years. Also here, the buffer storage takes over an important role, being able to mitigate permanent alternation of load and consequently suboptimal operation conditions - at the cost of efficiency.
Also with the given size of the buffer, it could be shown, that absorption chillers can take over high shares of the overall cold production in office buildings. For such profiles it would be outrageous to use absorption chillers to cover the rated performance of the overall cold concept, as this load is required for a very limited period. Consequently absorption chillers to take over base load and compression chiller to take over peak load seem to be an appropriate setup.
The worse dynamic behavior of absorption chillers was confirmed in the example, due to the fact that fast, heavy changes in load needed to be compensated by a switching by of the compression chiller and couldn't be taken over by the absorption chiller itself (if the buffer storage was not sufficient).

6 Economic evaluation

In this chapter the economic efficiency is assessed and compared to the currently in market dominant compression chiller. The evaluation is based on a scenario and cold concepts that were carved out under consideration of the insights that have been given previously. The method that is used in order to calculate the costs for cold generation is based on the guideline 2067 of the Association of German Engineers, that deals with the economic efficiency of building installations in general. For the reason that individual boundary conditions of each single project can be of critical importance for the profitability of the system, it is not possible to make universally valid statements. So the results that are presented in this chapter solely represent a certain selection. The obtained results and a sensitivity study make it possible do deprive tendencies and dependencies on certain parameters. In order to be able to perform fast, preliminary estimations of the profitability of a certain project in the future, an analysis tool was developed to calculate cold production costs for various different cold supplying systems and load profiles.

6.1 Costs

For the assessment of the economic efficiency of cooling systems all costs, that occur within the period under observation, have to be gathered and included in the calculation. In the VDI 2067 guideline the costs are divided into four main cost groups:

- Capital-related costs
- Demand-related costs
- Operation-related costs
- Other costs

6.1.1 Capital-related costs

Capital-related costs represent the expenditures for the initial investments for installation systems and the building components as well as replacements within the period of observation.

Capital-related costs for compression systems For the determination of the capital-related costs of compression systems (C_{CM}), a cost function that has been established by [39] is considered.

$$C_{CM}[€/kW_{Cold}] = (a \cdot Q_0[kW]^b + c) \cdot f_C \tag{6.1}$$

With this function both the costs for individual compression machines as well as for the whole compression cooling system can be estimated. The respective coefficients for the function are shown in table

6.1. The function itself was first derived in 2002 and is adapted to the current time by the application of the price index f_C for the respective years which is taken from [40]. The ones that are lodged for the "cooling system" include the following components:

- Compression chiller
- Cold- and cooling water cycle
- Water conditioning system
- Heat rejection system
- Measurement control and regulation technology

Table 6.1: Coefficients for equation6.1(values from [39])

Coefficients	Compression chiller	Cooling system
a=	4732.2487	4991.3436
b=	-0.7382	-0.6794
c=	109.30	179.63
f_C=	1,38	1,38

Capital-related costs for absorption systems For the calculation of investment costs for absorption chillers (C_{AM}) in general a cost function by [39] serves as reference. It does not include a heat rejection system and is given by:

$$C_{AM}[€/kW_{Cold}] = (14740.2095 \cdot Q_0[kW]^{-0,6849} + 3,29) \cdot f_C \tag{6.2}$$

It is valid for machines within a performance range of 50-4750kW and is also adjusted to the presence by the application of the price index f_C, which is shown in figure 6.1. Furthermore, the analyis tool provides the possibility to specify prices that are stated by manufacturers directly.

Capital-related costs for evaporative coolers For the calculation of capital-related costs of evaporative coolers (C_{CT}), basically two functions serve as data basis and can be chosen in the tool. On the one hand a cost function of [39], that is adapted to the present time by application of the price index for the respective years[40], is stored (equation 6.3).

$$C_{CT}[€/kW_{Recool}] = (2348,2 \cdot Q_{Recool}[kW]^{-1,0398} + 26,15) \cdot 1,34 \tag{6.3}$$

It is valid up to cooling performances of 1200kW. Furthermore an approximation function 6.4 has been calculated in excel. Therefore price indications of two companies that were collected in the course of the manufacturer inquiry were combined to a single function.

$$C_{CT}[€/kW_{Recool}] = 1922.7 \cdot Q_{Recool}[kW]^{-0.466} \cdot Q_{Recool}[kW] \tag{6.4}$$

The specific investment costs of evaporative coolers are shown in figure 6.1. It can be seen, that the costs of equation 6.4 (yellow line) are remarkably higher as the ones calculated by function 6.3 (red line). Reasons for this could be the low heat rejection temperatures and the consequently necessary big dimensioning of the heat rejection systems, that were requested in the enquiries. Furthermore the indications of the companies include sophisticated water treatment components such as water purification plant, water filtration system and water softening system.

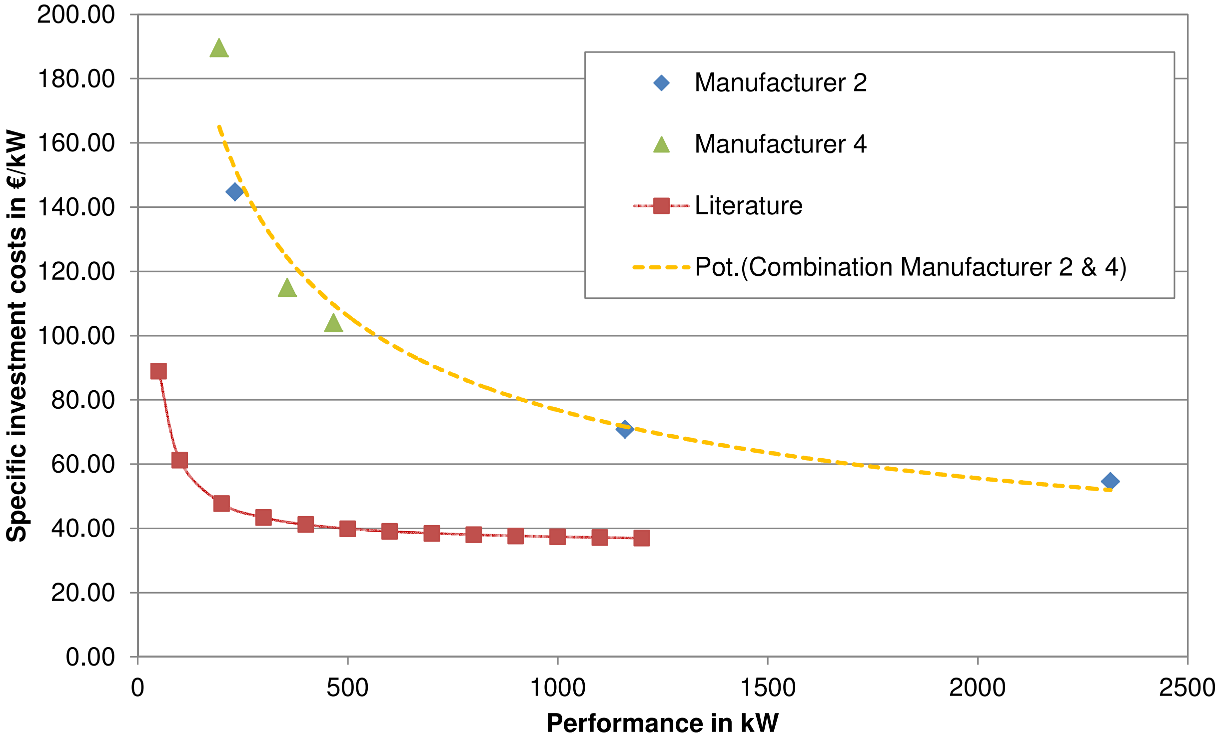

Figure 6.1: Specific costs for evaporative coolers

Capital-related costs for buffer storage systems The investment costs for buffer storage systems (C_{BS}) are calculated by a cost function that has been created as mean value of the quotations of two manufacturers (Lorenz[41] and Altmayer[42]).

$$C_{BS}[€/l] = 33.979 \cdot V_{BS}^{-0.373} \tag{6.5}$$

Capital-related costs for construction measurements Construction measurements are taken into account, if it is not possible to place system periphery in the building that is to be climatized. In this case, costs that would arise if a simple concrete housing would be built in order to store the machines are estimated. Expenditures for this simple housing have been estimated by an internal specialist, the values are shown in table 6.1.1 and consigned in the tool.

Table 6.2: Estimated construction costs for concrete housings

Size	Costs
$\leq 50\text{m}^2$	200 $€/m^2$
$\geq 50\text{m}^2$	150 $€/m^2$

6.1.2 Demand-related costs

Those expenditures are represented by the costs of the driving energy of the system, meaning for compression systems the costs for electricity and for absorption machines costs for heat. Furthermore, expenses for auxiliary electric energy (needed for the operation of pumps and heat rejection systems) and costs for the water that is consumed in the cooling tower are considered. The actual demand of the consumables cannot be determined exactly in advance but needs to be appraised/forecasted for the future. In [7] and [43] estimations about the demand for water (consisting of evaporation losses, desludging losses and splash water losses) and electricity can be found for both absorption and compression systems (inclusive heat sink system). The amount of driving energy for the different technologies can be calculated by knowledge of the COP of the respective systems. The underlying values are shown in table 6.3.

Table 6.3: Approximated energy demand of cooling systems

	Compression chiller	Absorption chiller
Driving energy	1/COP kWh$_{el}$ / kWh$_{Cold}$	1/COP kWh$_{th}$ / kWh$_{Cold}$
El. auxiliary energy	0.04 kWh$_{el}$ / kWh$_{Cold}$	0.07 kWh$_{th}$ / kWh$_{Cold}$
Water consumption	3.2 m^3/MWh$_{Cold}$	6.3 m^3/MWh$_{Cold}$

6.1.3 Operation-related costs

The operation-related costs include the expenses for maintenance and operation of the installations. Maintenance includes costs for servicing, inspection, repair and troubleshooting. The values for maintenance and operation are given for different systems in VDI2067 [44] and [39]. Both data sources can be chosen for the calculation in the program, whereby the data given by [44] are used as standard.

Table 6.4: Operation-related costs - based on investment costs

	Maintenance / servicing (VDI 2067) [44]	Maintenance / servicing [39]
Absorption chiller	3%	1%
Compression chiller	3,5%	4%
Heat rejection system	3,5%	-

6.1.4 Other costs

This cost group usually includes planning costs, insurance, taxes, general levies and charges profits and losses as well as costs for demolition and disposal. In the presented program costs for land/space leasing are considered as well as expenditures for the insurance of the machines. Table 6.5 shows the underlying values.

Table 6.5: Basic assumptions underlying the cost group "other costs"

Annual rent (land/space)	10 €/m^2 (estimated average value for Stuttgart)
Insurance	1% of investment costs

6.2 Investment appraisal

The calculation method that is used and described in the VDI guideline is the annuity method, which is a dynamic investment appraisal. It converts both on-off investments and ongoing payments during a chosen period of observation into a constant periodical payment - an annual installment.
Annuities are thereby calculated for each cost group by the help of annuity factors and - in the case of

changes in the ongoing costs during the period of observation - cash value factors. The formulas that are thereby used are given by [44]:

Annuity for the capital-related costs:

$$A_{N,C} = (A_0 - R_W) \cdot a \tag{6.6}$$

with:

- A_0 being the initial investment amount
- R_W being the residual value of the installation
- a being the annuity factor
- T_N being the number of years of the depreciation period
- T being the number of years of the observation period
- WACC being the weighted average costs of capital used as the interest factor

$$a = \frac{WACC - 1}{1 - WACC^{-T}} \tag{6.7}$$

$$R_W = A_0 \cdot \frac{T_N - T}{T_N} \cdot \frac{1}{WACC^T} \tag{6.8}$$

Annuity for demand-related costs:

$$A_{N,D} = A_{D1} \cdot a \cdot b_D \tag{6.9}$$

with:

- A_{D1} being the demand-related costs in the first year
- b_D being the price dynamic cash value factor for demand-related costs

Annuity for operation-related costs:

$$A_{N,Op} = A_{Op1} \cdot a \cdot b_{Op} \tag{6.10}$$

with:

- A_{Op1} being the operation-related costs in the first year
- b_{Op} being the price dynamic cash value for operation-related costs

Annuity for other costs:

$$A_{N,O} = A_{O1} \cdot a \cdot b_O \tag{6.11}$$

with

- A_{O1} being the other costs in the first year
- b_O being the price dynamic cash value for other costs

The price dynamic cash values are calculated by applying the respective price indices (r) in the following formula:

$$b = \frac{1 - (\frac{r}{WACC^T})}{WACC - r} \tag{6.12}$$

Overall annuity:

The individual calculated annuities are summarized to an overall annuity.

$$A_N = A_{N,C} + A_{N,D} + A_{N,Op} + A_{N,O} \tag{6.13}$$

By the division of the annuity by the expected yearly cold production, the specific costs for cold generation (p_{Cold}) are calculated.

$$p_{Cold}[€/kWh] = \frac{A_N}{P_{Cold,a}[kWh/a]} \tag{6.14}$$

6.3 Computational model

In the following the computational model that is used for the economic assessment of different cold supplying concepts is introduced. Figure 6.2 gives an overview over the different settings that can be made for the calculation as well as over the functions and data that are stored in the program.

The basic selection options are firstly related to the cold demand. Thereby the maximum required cold performance of the system as well as the annual distribution of the cold demand has to be determined. This can either be done by specification of an annual load curve or in a simplified case by determination of the amount of full load hours of the cooling system. As a second step the method of cold production must be specified by determination of the cold generation concept. Hereby it is possible to perform the calculations for an individual technology as well as for a combination of compression and absorption chillers in all different ways. It can be individually chosen to what extend the cold is provided by the individual technology. Furthermore the share of the provision of cold by the absorption chiller can be calculated under simplifying assumptions, given that an annual load curve is stored. The same applies for the annual effective COP of the absorption chiller, which can either be specified directly or calculated by super-imposition of a selected partial load curve and the annual load duration curve. The cold generation system contains in each case - no matter if compression and/or absorption chiller - a certain buffer capacity. This is a result of the indications given by the manufacturers of absorption chillers as well as the experience in praxis (see chapter 5) that clearly recommend an integration of such a capacity. The dimensioning of the storage for absorption chillers is thereby based on the mean value of the indications given by the manufacturers Man3, Man4 (cf. chapter4.2), while for compression chillers, the method of [20] is used (cf. chapter 2.5.2).

In order to be able to conduct the calculation, the program has stored the different cost functions that have been introduced in chapter 6.1. As for the different types of chillers and the cooling towers different sources are available, it is possible to choose between the different sources for the respective contemplation. Furthermore the excel tool stores partial load curves for different absorption chillers (see figure 4.3),indications about the consumption of different forms of energy and labour (see table 6.3), as well as functions for the space that is required in order to place the system (see appendix A.4). In addition prices for the different consumables as well as the respective price indices are lodged although those can be verified individually as well as the $WACC$, the marge and the period under observation for which

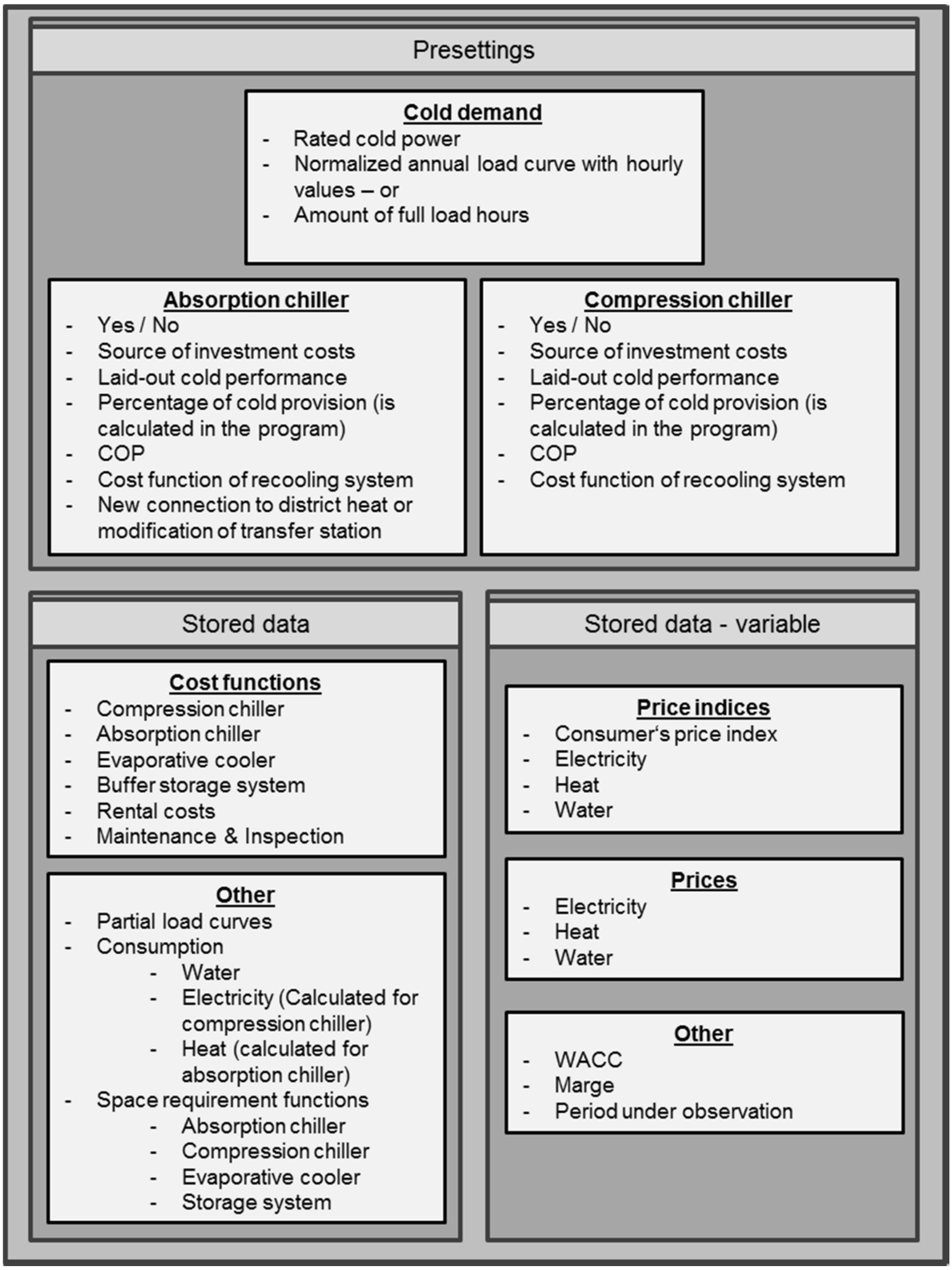

Figure 6.2: Overview over possible selection possibilities as well as stored data and functions

the calculation is conducted.

As already mentioned, the model provides the cost for cold production. The subsequent distribution of the cold within the building and thereby the associated costs for the respective infrastructure are not part of the contemplation and consequently not considered. Boundary conditions such as required cold water, heat source and recooling temperatures do not directly enter the calculation but indirectly by indication of the COP of the machine.

6.4 Base scenario and cold concepts

In the following the base scenario and the cold concepts for which the economic evaluation in this work is conducted, are presented.

Base scenario

The assumptions that are taken in the base scenario are related to prices, price indices, the choice of time intervals, the determination of efficiency of the components as well as sources for the costs of the components. In table 6.6 and 6.7 these assumptions are summarized. The price for electricity in the first year of the contemplation is derived from [45] and is the mean value of the trading and industry electricity price in 2015. The price index is approximated, so that the development is similar to the prediction of the development of the electricity price in the reference model of [45]. The price of heat is chosen to a value that enables a sensible comparison to compression machines. The respective price index is the same as for electricity. The price for fresh water in the first year is the mean value for drinking water in 2014 in Baden-Württemberg [46], and develops with the consumer's price index, which is the average value of the recent years taken from [47]. The same price index is used for waste water. The respective price for waste water in the first year is applied on the average price for waste water in Baden-Württemberg given in [46]. Whilst costs for fresh water arise for the entire consumed water, expenditures for waste water are just charged for the desludging losses (being determined to 50% of the overall water demand [7]).

Table 6.6: Underlying prices for energy carriers

	Price	**Price index**
electricity	17 ct/kWh	3%
fresh water	2 €/m^3	1.625%
waste water	2 €/m^3	1.625%
heat	15 €/MWh	3%

With regard to the observation period (T), the recommendation of the VDI 2067 guideline is taken over, being 15 years. The same applies to the depreciation period (T_N), being 18 years for the absorption chiller, and 15 respectively 20 years for the compression chiller and the recooling system. The $WACC$, which is used as interest factor, is the average value of the energy and power generation sector in Germany, determined in [48].
The investment costs of both, the absorption chiller as well as the compression chiller, are calculated with the cost function from literature (equation 6.2, equation 6.1 (coefficients for the compression chiller)). The investment costs of the heat sink system are calculated by equation 6.4.
The effective COP of the compression chiller amounts to 4, whilst the rated COP of the absorption chiller is 0.75, being adapted by the partial load curve from literature for the respective cold concepts (cf. **??**).

This is done by superimposition of the partial load curve and the annual load profile. The annual load curve underlying the base scenario is the one of the company being active in the medical sector (cf. figure 3.4).

Table 6.7: Settings of the base scenario

Consumer's price index	1,625 %
T_N absorption chiller	18 a [44]
T_N compression chiller	15 a [44]
T_N recooling system	20 a [44]
T	15 a [44]
WACC	6,4% (mean value in energy sector [48])
Cost function absorption chiller	Function from [39]
Cost function compression chiller	Function from [39] (solely chiller)
Cost function evaporative cooler	Own function
Partial load behavior absorption chiller	Function from literature [11]
Effective COP compression chiller Effective COP absorption chiller	4 Adapted[1] - underlying rated value: 0.75

Cold generation concepts

In order to keep a clear overview in the consequent evaluation, the number of different cold generation concepts (in the following also called cold concepts) that are assessed is limited to six. On the one hand the single technologies - compression chiller and absorption chiller - are assessed, being represented by concept 1 and 2. On the other hand, combinations of the two types are represented by concept 3 and concept 4. In each of these concepts, the absorption chiller has a rated performance of 40% of the maximum performance of the overall system.

Table 6.8: Cold generation concepts

Cold concept	Absorption chiller			Compression chiller		
	YES/NO	$Q_{0,rated,AC}$ / $Q_{0,rated,System}$	Share of annual cold provision	YES/NO	$Q_{0,rated,CC}$ / $Q_{0,rated,System}$	Share of annual cold provision
1	NO	0	0%	YES	1	100%
2	YES	1	100%	NO	0	0%
3a	YES	0.4	76%	YES	0.6	24%
3b	YES	0.4	53%	YES	0.6	47%
4a	YES	0.4	76%	YES	1	24%
4b	YES	0.4	53%	YES	1	47%

Under review of the load profiles and daily courses of the cold demand presented in chapter 3, this value is freely chosen but to a value which seems to be adequate in order to guarantee a secure provision of base load by the absorber. This security of supply is increased by a certain buffer storage capacity which is included in each cold concept, being calculated for each technology separately. In cold concepts 3a,3b the perfomance of the compression chiller complements the performance of the absorption chiller, so that the overall required maximum performance is reached. In concept 4a,4b the compression chiller is dimensioned in order to be able to take over the entire required maximum performance and the sum of both exceeds consequently this value. In concept 3a and 4a, the absorption chiller takes over the complete base load within its performance range. In contrast in cold concept 3b and 4b, the absorption

[1]The calculation is done by super-imposition of a selected partial load curve and the annual load duration curve of the cooling demand

chiller solely takes over the base load in the warmer months (in which a lack of district heat demand can be observed) and the compression chiller respectively takes over base load operation in the cold months (November, December, January, February and March). These two possibilities (a and b) of cold provision are depicted in figure 6.3.

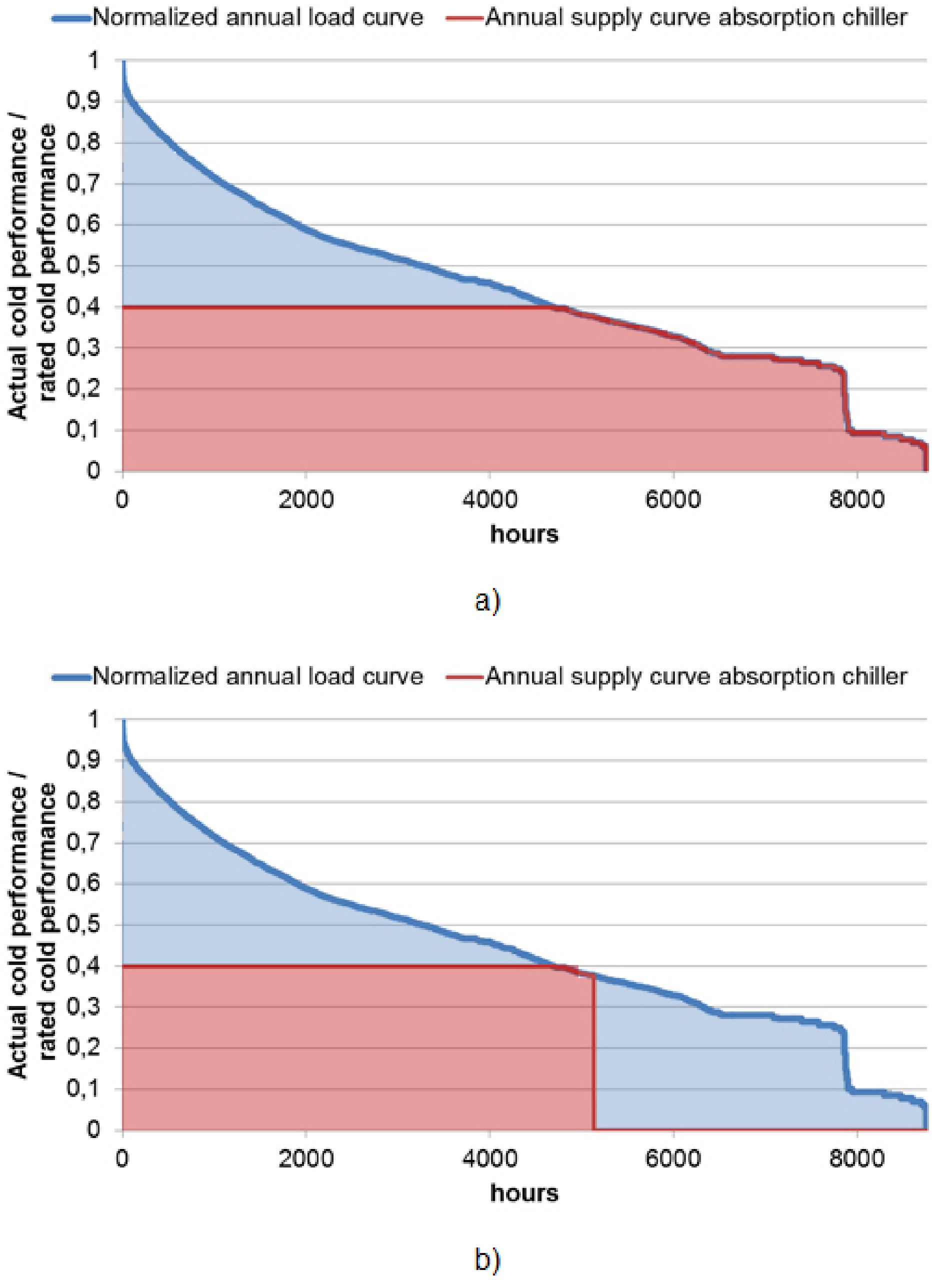

Figure 6.3: Provision of cold, underlying concept 3a,3b,4a and 4b (cf. table 6.8)

6.5 Evaluation

6.5.1 Critical prices for heat

Figure 6.4 shows the production costs of cold that are calculated for the 6 different cold provision concepts in dependency of the price for heat. The underlying maximum cold performance - which would be in this case the contractually guaranteed cold performance - of the presented data amounts to 1000kW. The annual cold production amounts to 3873.8 MWh, being in case of concept 1 and 2 100% provided by compression chiller respectively absorption chiller, in case of 3a and 4a to 76% and in case of 3b and 4b to 53% by the absorption chiller.

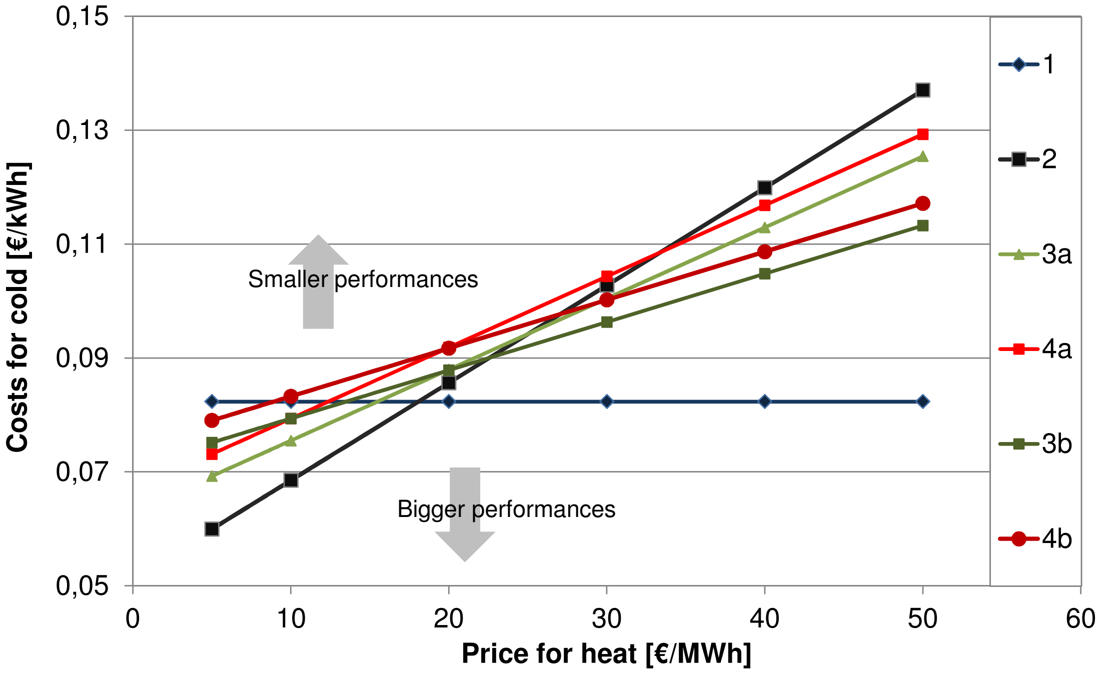

Figure 6.4: Costs for cold in dependence of the price for heat (underlying the base scenario cf. table 6.7 and the cold concepts given in table 6.8)

For the assumptions of the base scenario, the entirely generation of cold by compression chillers (concept 1) exhibits a specific price of about 8.2ct/kWh. It is the only concept that is completely independent of the price for heat as no thermal driven component is installed. For the other concepts a linear dependency on the price of heat can be observed and consequently can be expressed by a linear function with a positive pitch. The pure absorption chiller exhibits thereby the biggest dependency on the heat price, inducible by the biggest pitch. Nevertheless the critical heat price, meaning the value when it matches the price for cold of concept 1, is the biggest for concept 2 and amounts to about 18.1€/MWh. This means that if heat cheaper than 18.1€/MWh would be available, this concept would be the cheapest option. Concept 3a and 4a exhibit the second biggest dependency on the heat price, because, compared to 3b and 4b, a bigger share of cold is provided by absorption chillers. The price of heat, for which the respective concepts becomes favourable, are thereby 16.5€/MWh respectively 12.8€/MWh. For concept 3b the critical price amounts to 13.5€/MWh and for concept 4b 9.0€/MWh. Table 6.9 shows an overview over the critical heat prices for the different concepts and different maximum cold performances. Graphically a decrease of cold performance shifts the lines in diagram 6.4 upwards, whilst an increase leads to a shift in downward direction.

Table 6.9: Critical prices for heat (matching the price of an entirely generation of cold by compression chillers). Results for the different cold supplying concepts and different rated performances (values in €/MWh).

Max. performance	Concept 2	Concept 3a	Concept 4a	Concept 3b	Concept 4b
500 kW	15.3	11.8	8.2	7.5	2.6
1000 kW	18.1	16.5	12.8	13.5	9.0
2000 kW	20.1	18.5	15.6	16.8	13.0

Apart from the exclusive consideration of the cold generation costs, it must be considered, that the different concepts imply different qualities of cold provision. In comparison to concept 1 and 2, the other cold generation concepts provide additional security of supply. If one of the systems has a malfunction, it is possible to replace it by the other system within its capabilities. Concept 4 provides hereby the highest level of security as the compression chiller is dimensioned big enough to cover the overall demand at any time. This is of particular importance if the field of application has very high demands on the security of supply, like it is the case e.g. for data centers. In such facilities the need for redundant systems is quite high and concepts like "concept 4" are more recommendable. Furthermore it must be considered, that especially concept 2 is very unlikely to be reasonable in reality, due to the bad dynamic behavior of absorption machines. In this case peaks couldn't be taken over by switching by of compression chiller capacity, but would need to be taken over completely by the storage system, which therefore should be dimensioned far greater than conventionally done.

6.5.2 Composition of cold production costs

Figure 6.5 shows the composition of the cold production costs for the single technologies and two different performance sizes (100kW and 1000kW), being valid for the assumptions of the base scenario. The annual cold production underlying the contemplation amounts to 3000 full load hours and the annual average COP of the compression and absorption chiller amounts to 4 respectively to 0.7.
First of all it can be noticed that the costs for cold production of the absorption chiller exceeds the respective expenditures of the compression chiller for the smaller performance remarkably (about 6ct/kWh). For the higher performance however the difference between both technologies is significantly reduced and the absorption chiller is even the less expansive option.

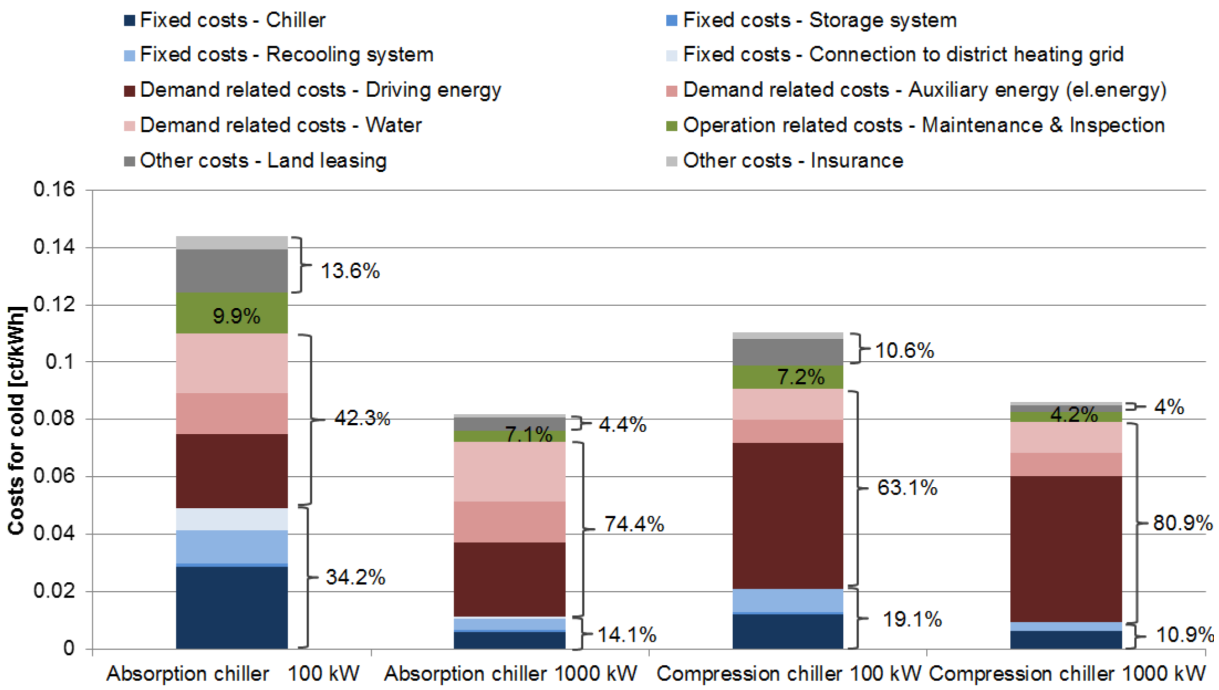

Figure 6.5: Composition of the costs for cold

Fixed costs

The difference in the initial investment costs between absorption and compression chillers can be well observed for the small performance class. The investment costs of the absorption machine are hereby more than twice as high as for the compression chiller. For both refrigerator types, the share and also the costs per produced cold of the fixed costs decrease significantly with increasing cold performance. This effect is much more pronounced for the absorption chiller, because the fixed costs share of the compression chiller is already considerably smaller for the smaller cooling performance, so that there is less potential for reduction. Consequently the decrease in investment costs of the compression chiller, that is associated to an increase of cold performance, does not greatly affect the overall cold production costs. On the other hand nearly a saving of 3ct/kWh can be observed for absorption systems. The chiller clearly represents the biggest part of the investment costs, while for compression systems the costs for the recooling system are similar to those of the refrigerator. The differences for the costs of the recooling system between both technologies can be traced back to the higher recooling capacity that is required for the absorption chiller. This is due to the high amount of anergy that is supplied in such systems and consequently has to be removed in the heat sink system in order to guarantee stationary conditions.

Other costs and operation related costs

Similar as for fixed costs, an increase of cold performance reduces the specific costs for insurance, land leasing as well as the operation related costs. This leads for both technologies to a significant reduction of the price if the performance of the system is increased. Comparing the two technologies, both, the operation related costs as well as the other costs, are higher for absorption systems. The difference between the systems disappears the bigger the performance gets.

Demand related costs

The demand related costs are obviously the dominating cost group for compression chillers. Even for rather small cold performances - like in this case 100kW - the share of the demand related costs is about 63.1%. For 1000kW the share even increases to 80.9% of the overall cold production costs. The demand related expenditures for absorption chillers are lower compared to the compression chillers. This difference is mainly accountable to the significantly lower costs for the main driving energy heat compared to the expenses that arise for the operation of the compression chiller with electric energy. The costs for water and auxiliary energy are in fact higher for absorption machines and nearly as high as the expenses for the driving energy. For both systems it is valid, that the demand related costs are not dependent on the size (performance) of the machine. Those costs are solely dependent on the operating parameters (consumption of the machines expressed by the COPs of the chillers and the other components such as pumps and recooling systems), as well as the prices for energy, energy carriers and consumables.

Consequently it can be stated that absorption chillers usually become more interesting, the bigger the system performance is. This is because the influence of the higher fixed costs as well as operation-related and other costs decrease with increasing cold demand. If instead of the maximum cold performance, the amount of full load hours would be increased, the same effect could be observed. The absolute value of the fixed costs would remain the same, but since the costs are related to the yearly cold performance, this specific value decreases.

6.5.3 Sensitivity analysis

For the calculations presented above, assumptions were made and solely a vanishing small number of options could be covered. For the reason that these taken assumptions are not universally valid, sensitivities of the costs for cold generation are assessed by variation of different influencing factors. The investigation is thereby conducted for each technology individually as this is considered to be the best possibility to deduce trends.

In figure 6.6 and 6.7 the results of this contemplation are shown. Starting from the base scenario, that matches the constraints of the preceding considerations and a rated capacity of 500kW, single parameters are varied in two directions - meaning the single values are decreased up to 100% in case of negative values and respectively increased for positive values. Table 6.10 shows these initial values as well as the respective maximum and minimum values.

Table 6.10: Overview over parameter variations

Deviation	-100%	0%	100%
Price for electricity	0 ct/kWh	17 ct/kWh	34 ct/kWh
Price for heat	0 ct/kWh	1.5 ct/kWh	3 ct/kWh
Heat price index	0%	3%	6%
Electricity price index	0%	3%	6%
Full-load hours	0	3000	6000
Rated performance	0 kW	500 kW	1000 kW
$WACC$	0%	6.4%	12.8%
COP_{CM}/COP_{AM}	0/0	4/0.7	8/-

Absorption chiller The biggest deviations of the costs of cold generation are represented by the curves that illustrate the changes of the COP respectively the number of full-load hours, which almost proceed on the same line. A negative exponential behavior can be observed. This means, that small deviations exert less heavily influence on the costs (e.g. a decrease of the working load of about 20% leads to an increase of the costs from about 0.092€/kWh to roughly 0.099€/kWh), while a big decrease e.g. by the factor 0.8 leads to an increase of costs to about 0.215€/kWh. This means that the costs are more than doubled. While such a decrease is not realistic for the COP of the chillers, such low working loads can occur if the systems are used in the wrong applications. An increase of the named parameters lowers the costs for cold production. However it must be acknowledged that this is a purely hypothetical consideration for the COP, as an effective COP higher than 0.75-0.8 is highly unrealistic and a value higher than 1 even impossible for single-stage absorption chillers. For the number of full-load hours even an increase of 100% meaning up to 6000 full-load hours would theoretically be possible and would lead to costs of about 0.076€/kWh. Consequently this contemplation emphasizes again which decisive role the load profiles play on the economic performance of absorption units.

In addition to the COP and the working load, the rated power and therefore the size of the absorption chiller, shows a negative exponential relationship and has a high influence on the costs of cold. This could be already shown in chapter 6.5.2. This means that a decrease of the performance leads to an increase in the costs for cold, being higher, the smaller the machines get and leads to lower costs, the bigger the machines get. The savings which can be reached here are however less distinctive due to the demand-related costs, which are not dependent on the size of the machine. Though a doubling of the cold performance to 1000kW leads solely to a reduction of costs of about 1ct/kWh.

A change in the development of the prices for electricity and heat does not decisively affect the costs for cold production. If the heat price index is doubled, the costs for cold are increased to about 0.097€/kWh,

while a doubling of the electricity price index leads to an increase of about 3% to roughly 0.05€/kWh. If the factors are decreased, the costs are lowered as well, although this variation is even less pronounced than the increase that is observed in other direction.
Similar to the price indices, a variation of the $WACC$ leads to rather small deviations of the costs for cold, increasing/decreasing with an increase/decrease of the parameter.
The remaining two parameters that are considered in this sensitivity study, exhibit a linear dependency between the costs for cold and the respective parameters. Both of them have a positive slope meaning that with increasing prices for heat or electricity, the costs for cold production increase. The dependency on the price for heat is thereby higher, being expressed by the bigger slope. Hereby it must be considered however, that a variation of the price for heat of about 20% means a variation of 6€/kWh while the same variation of the electricity price would mean a decrease/increase of about 34€/kWh. This means that the slope of the line is dependent on the choice of the initial price and would be even steeper if the initial price for heat would have been bigger. If the electricity would be for free, the costs for cold would still amount to about 0.077€/kWh. If heat would be available for free, costs of about 0.066€/kWh would arise considering the given boundary conditions.

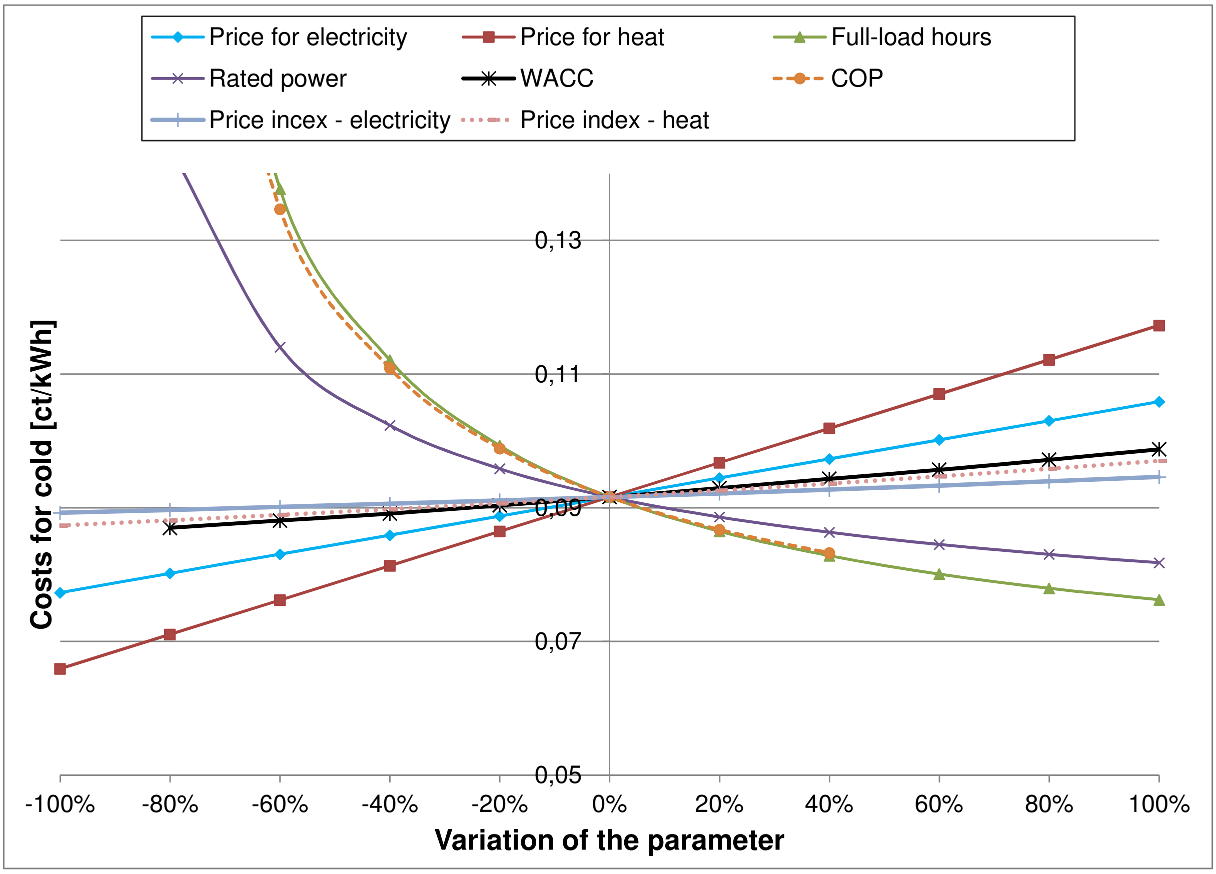

Figure 6.6: Influence of certain parameters on the costs of cold production by absorption chillers (cf. table 6.10)

Compression chiller Similar to the absorption chiller, the compression chiller shows a strong, negative exponential dependency on the number of full-load hours and in particular on the COP of the system. If the COP is halved to a value of 2, the costs for cold are almost doubled. On the other hand an increase of 50% to a value of 6 leads to a decrease of costs to about 0.072€/kWh. The compression chiller reacts significantly less sensitive on the variation of the rated power. Only for bigger deviations in negative direction - meaning a strong reduction of rated power - a higher increase in costs can be observed.

Particularly pronounced is the linear dependency on the price for electricity. If the electricity price would be doubled, the costs for cold production would increase by almost 66% from 0.090€/kWh to 0.149€/kWh. A variation of the parameter in opposite direction would lead to costs for cold of only 0.031€/kWh. This behavior can be explained by the high share of the demand-related costs for electricity on the overall costs for cold production (see chapter 6.5.2). The variation of the electricity price index does not lead to such distinctive deviations of the costs for cold, nevertheless it shows a higher dependency compared to the absorption chiller, which is due to the higher electricity requirements of the machine. A doubling of the price index to about 6% - which is the approximately value of the price index of the recent years in Germany [45] - leads to costs for cold of about 0.102€/kWh. The price for heat as well as the price development for heat does obviously not affect the costs for cold for compression systems, due to the fact that heat is not required. With regard to the WACC, variations are less pronounced compared to the absorption chiller and are generally small.

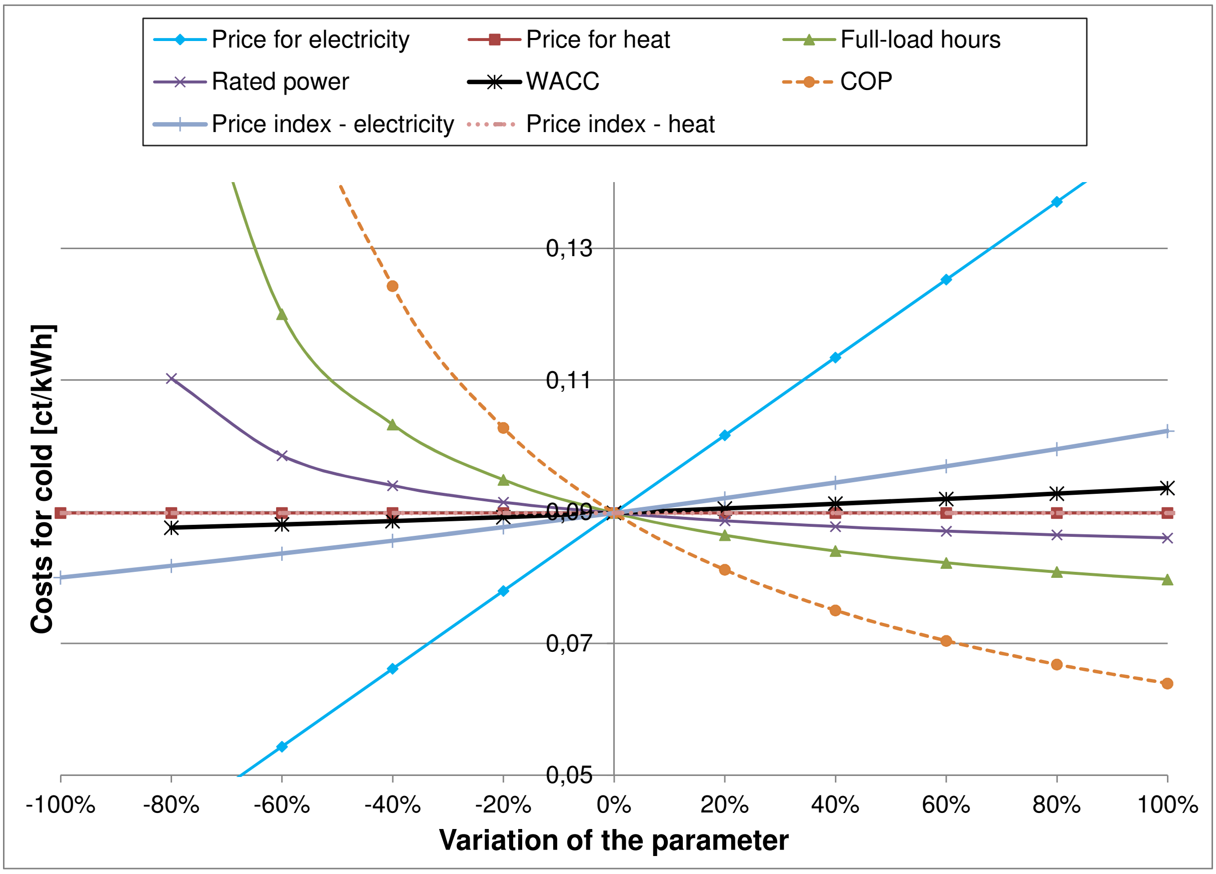

Figure 6.7: Influence of certain parameters on the costs of cold production by compression chillers (cf. table 6.10)

6.6 Conclusion

The economic analysis confirms the assumption, that compression systems are significantly cheaper for applications, that have small cold demands. This is mainly related to the significantly higher investment costs for small absorption machines. For bigger cold demands, this difference in investment costs vanishes and the demand-related costs become the dominating cost group. These costs are strongly dependent on the efficiency of the processes and are not dependent on the performance of the system. For the underlying base scenario, the absorption system exhibits lower demand-related costs. The most crucial factors for the cost effectiveness of absorption chillers are the COP and the load factor (expressed

by the number of full-load hours), as well as the rated performance and price for heat.
The critical price for heat for which absorption chillers are competitive to compression chillers is in a range between 2€/MWh) and 20€/MWh for the considered scenarios and cold concepts. This price is thereby far lower than the average price for district heat, which is about 50-80€/MWh in Germany [49]. For a consumer it is therefore hardly conceivable, to run its own absorption chiller with heat, purchased by a district heating company to regular conditions. For a utility on the other hand it is hardly deliverable to sell the same product to distinctly different prices. For this reason, the only practicable solution are contracting models, by which the utility is owner of the cold supplying system on-site and sells the product cold to its costumers.

7 Environmental indicators

In a final step the environmental footprint of the two technologies is determined. This is done by the estimation of the specific primary energy demand, that is associated with the consumption of electricity and/or heat during cold production. Furthermore, the TEWI (cf. chapter 2.15) of compression and absorption machines is determined and thereby the equivalent specific CO_2 emissions of the technologies are calculated.

7.1 Primary energy factors

By the help of primary energy factors (f_p), the consumption of primary energy, which is associated with the production of end energy, can be determined. The factor is defined as quotient between primary energy used - including upstream processes such as extraction, treatment and transport of the energy carriers - and the calorific value of the produced energy source [**?**]. Table 7.1 shows the primary energy (f_p) and also the CO_2 equivalency factors (β), that are used in the consequent calculations. The primary energy factor, given for district heat is the certified published value of EnBW's district heating grids [50]. For electricity the official value of the German electricity mix, which is valid from the year 2016, is taken. This value is given in the Energy Saving Ordinance [51]. It was corrected from 2.6 to 2.4 in 2014 and will be lowered to 1.8 with the start of 2016. This decrease is associated with the increasing share of renewables in the German electricity mix.

Table 7.1: Primary energy and CO_2 equivalency factors

	District heat	**German electricity mix**
Primary energy factor	f_p = 0.5484 kWh$_{prim}$/kWh$_{end}$	f_p =1.8 kWh$_{prim}$/kWh$_{end}$
CO_2 equivalency factor	β = 150 g$_{CO_2}$/kWh$_{end}$	β = 617 g$_{CO_2}$/kWh$_{end}$

7.2 CO_2 equivalency factors

CO_2 equivalency factors (f_{CO_2}) describe the amount of CO_2, that would be equivalent to the amount of greenhouse gases that is emitted by operation of a certain technology. The CO_2 equivalency factor for electricity is taken from [52] and is about 617 g/kWh. The value for district heat is set to 150 g/kWh. This factor is chosen as a realistic value for a district heating grid, that has a high share of CHP. Among them, the majority of the plants are coal-fired power plants and a smaller amount of gas-fired power plants.
The TEWI concept distinguishes between indirect and direct emissions. In the following, the indirect CO_2 emissions are calculated by multiplication of the consumed energy (electricity and/or heat) with the respective equivalency factors. Water - used as refrigerant for absorption chillers - does not have a global warming potential, so that there are no direct emissions considered. According to its definition (see chapter **??**), the direct emissions of compression chillers are dependent on its leakage ratio, refrigerant

charge and type of refrigerant. For the reason that these options are so various, it is difficult to select specific cases and it would not be conducive. Furthermore it is just intended to give estimations of these emissions. As a consequence, the direct emissions of compression chillers are set to be 10% of the indirect emissions, being generally a sensible value (according to indications given in [53]).

Evaluation

The results of the CO_2 emissions and primary energy demand of the different cold concepts (cf. table 6.8) are presented in table 7.2. The results are underlying the assumptions of the base scenario, considering COPs of absorption/compression machines of 0.75 respectively 4. When comparing the single technologies, it can be seen, that the absorption chiller (concept2) implies higher primary energy demands and specific CO_2 emissions than the compression chiller. The consumption of the different energy sources is consequently too high in order to take advantage of the lower primary energy and CO_2 equivalency factors. In general, the specific emissions/energy demands of the different technologies is solely dependent on the factors (f_p, β) and the consumption of energy (heat and/or electricity)
The other four cold concepts 3a,3b and 4a,4b are levelling of itself in between the two single technologies. The higher the share of absorption chillers thereby is, the higher are the specific emissions/energy demands.
Though absorption chillers exhibits worse behaviour with regard to the two observed values, the use of absorption chillers could still be beneficial in an overall accounting. In its proposed form, solely the cold generation side, but not potential effects on the general district heat supply are considered. An example could be, that new customers for district heat could be won, when both cold and heat supply could be offered in combination.

Table 7.2: Primary energy demand and specific CO_2 emissions for the concepts given in table 6.8

Concept	**1**	**2**	**3a**	**3b**	**4a**	**4b**	**unit**
Primary energy demand	0.52	0.86	0.78	0.70	0.78	0.70	$kWh_{prim}/kWhcold$
Specific CO_2 emissions	173.9	243.2	232.1	221.4	232.1	221.4	g_{CO_2}/kWh_{cold}

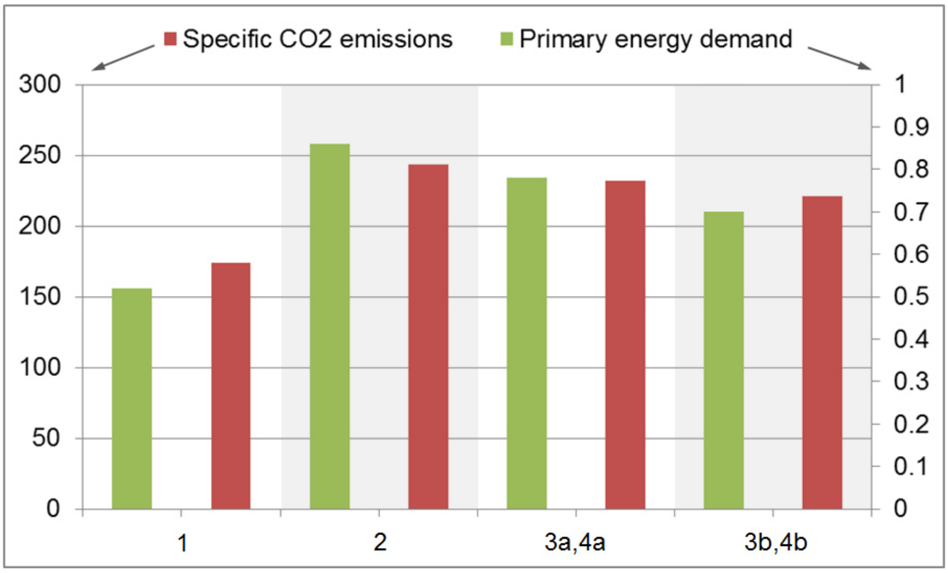

Figure 7.1: Specific CO_2 emissions [g/kWh] and primary energy demand [-] for the cold concepts presented in table 6.8

8 Summary and conclusion

In the present work, the currently dominant refrigeration technology, the compression chiller and the competitive absorption chiller are evaluated and compared. The main difference between both technologies is the different driving energy input. Whilst the compression chiller is electrically driven, the absorption chiller mainly uses thermal energy and needs only little electricity to run its pumps. Linked with this different energy source is the declaration of the efficiency of the technologies. In case of compression chillers, the energy efficiency ratio is defined by cold performance per supplied electric energy, for absorption chillers it is determined by the cold performance divided by the supplied heat (heat ratio). The values for compression chillers are usually in the range between 3 to 6 and thus exceed typical values for single stage absorption chillers by many time (typically between 0.6 and 0.8 in the case of LiBr-water). This is due to the fact, that the upstream process of generation of electricity e.g. in power plants is not considered in the contemplation of the efficiency of compressors. The electricity supplied to compression machines is pure exergy. Heat flow that drives the absorption chiller includes beside of its exergetic share a remarkable amount of anergy. As a consequence, absorption chillers require bigger heat rejection systems which increase the costs for operation and investment, as well as space requirements. With regard to dynamic operation conditions compression machines are advantageous, especially concerning peak load operations. Absorption machines are beneficial with respect to its partial load behaviour, but require in general a more constant load profile without heavy fluctuations. Compression machines cover the entire range of common cooling applications - from low temperature to air-conditioning applications. Absorption chillers can be divided into two groups, classified according to its working fluids. Water-ammonia systems are used in low temperature applications (T_0<6°C), whilst LiBr-water single-stage machines could be identified as the only reasonable technology in combination with the use of district heat due to the associated temperature of district heating grids. In summertime, when the need for cooling is particularly high, these temperatures are in a range of approximately 80°C. By the assessment of the industrial sector and the sector air-conditioning in buildings, branches with high cold demand can be ranked as less suitable for the application of the identified absorption machines. This is a result of the low cold temperatures, that are required e.g. in food, chemical, construction- and building- materials industry. However industrial sectors still remain as potential fields of application. Among them are automotive, process cooling, pharmaceutical industry and plastic and rubber processing. Furthermore the rapidly growing market of air-conditioning in office buildings represent a potential application field for absorption machines, especially when combined with server rooms, data centers and/or industrial cooling.

The absorption chiller market in Germany is dominated by foreign companies. By the help of German sales partners and specialists in HVAC, the machines are offered in Germany. A meaningful comparison of the different machines, by openly available indications of the individual manufacturers, is hardly possible, as the indications do not necessarily underly uniform assumptions and boundary conditions. Nevertheless, by consultation of manufacturers and retailers, such a comparability was reached for a certain selection of absorption chillers and certain specified constraints, that match the requirements for district heating grids. A final conclusion is however still challenging, because solely theoretical indications, given by manufacturers, can be evaluated. This would require the analysis of a sufficiently large number of field tests. The specific investment costs for the machines of the various manufacturers partially differ remarkably from each other and from the cost function, given in the literature. This is especially the case for smaller cold performances and decreases for bigger ones. Furthermore remarkably were the differences in the specification of the recommended buffer storage size.

The evaluation of a real cold supplying system of a municipal utility showed high agreement between theoretical specifications and real operating characteristics. Hence, it can be assumed, that the performance data provided by the respective manufacturer can be achieved in reality. Furthermore, the benefits of a constant base load operation of absorption chillers could be approved in the example. For the given office building, that includes server rooms, a high amount of cold could be provided by the absorption chiller. This share could even be increased, if the capacity of the buffer storage would be increased and thus the influence of the storage on the operational behavior and the load factor of the absorption chiller could be detected. In light of the above, indications of some manufacturers, regarding the size of the buffer storage, seem to be less appropriate in order to reach a possibly high share of cold provision of the absorption chiller. Furthermore the influence of spatial conditions on the success of absorption systems could be shown, inhibiting or restricting the application of absorption machines.

The economic analysis of the different technologies highlights the influence of the performance on the costs for cold generation. For small cooling performances, compression chillers seem significantly superior and an application of absorption chillers can hardly be recommended. With increasing performance class, this difference vanishes and absorption chillers can even be the cheaper solution. The critical prices for heat, for which absorption chillers become competitive to compression chillers, are significantly lower than the price that is typically paid for district heat. In this respect, the operation of absorption chillers is solely representable, if the consumer does not pay for the product "heat" in order to operate his own absorption machine, but for the product "cold". This can be offered in form of a contracting model, meaning that the costumer agrees on a contractually defined price per purchased amount of cold. As an additional benefit, the high investment costs are omitted for the consumer.

In general, it is not suggestive to compare the different technologies separately, due to the different operational characteristics. Absorption chillers should be used as base load machine and complemented by compression chillers, taking over the peak loads. Furthermore, a purely economical evaluation of the cold supplying systems is not advisable, because of different qualities that are implied by such systems. If a redundancy is required, absorption chillers represent a meaningful supplementation to compression chillers. Malfunctions of one system could be bridged by the respective other system, as well as supplying gaps of one of the driving energy sources. Necessary emergency power supplies could be significantly decreased due to the distinctly lower electricity requirements of absorption systems.

In this work, the economic and ecological evaluation solely considers the cold production side. In case of the absorption chiller, the district heat and the provision and generation of heat are exclusively represented by a single factor, the price for heat. In order to assess the applicability of this price and to identify the in fact meaningful operation periods of the absorption chiller, it would be necessary to model the district heating grid and to consider the scheduling of plants, as well as price signals for heat and electricity.

Analyzing environmental indicators, it can be stated, that the cold produced by absorption chillers causes more emissions than compression chillers. This is a result of the increased share of renewable energies in the electricity mix, leading to lower CO_2 equivalency factors (β) and primary energy demands (f_p). Though, in combination with a supply of heat by district heating grids, the overall ecological footprint might be better, due to the fact, that the primary energy demand as well as the specific CO_2 emissions of district heat is superior to those of decentral oil- or gas-fired heating systems.

Finally it can be said, that absorption systems may be a sensible solution for district heating companies in certain cases. A general recommendation for the use of absorption systems is however not possible. Boundary conditions of each single project need to be considered very carefully. The most important ones thereby can be given by: the price for heat, the heat source temperature that is available (for that the machines are laid-out possibly high), the available space, the required cold temperature, the load profile and the required cold performance.

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A Appendix

Table A.1: Technical specifications of Yazaki water-fired absorption chillers

Parameter	WFC-SC05	WFC-SC10	WFC-SC20	WFC-SC30	WFC-SC50
P [kW]	17.5	35	70	105	175
$T_{Coldwater,in}$ [°C]	12.5	12.5	12.5	12.5	12.5
$T_{Coldwater,out}$ [°C]	7	7	7	7	7
$T_{Hotwater,in}$ [°C]	88 (max.95, min.75)	88 (max.95, min.75)	88 (max.95, min.75)	88 (max.95, min.75)	88 (max.95, min.75)
$T_{Hotwater,out}$ [°C]	83	83	83	83	83
$T_{Coolingwater,in}$ [°C]	30	30	27	27	31.5
$T_{Coolingwater,out}$ [°C]	36	35	32	32	35
COP	0.7	0.7	0.7	0.7	0.7
Op. weight [t]	0.42	0.604	1.156	1.801	2.725
Power consumption [kW]	0.048-0.072	0.21	0.26	0.31	0.59
Dimensions [mm] length/width/height	744/594/ 1786	970/760/ 1920	1300/1060/ 2030	1545/1380/ 2065	2060/1785/ 2223

Table A.2: Technical specifications of World Energy's water-fired absorption chillers

Parameter	HWAR-L135	HWAR-L300	2AA135	2AA300	2AB135	2AB300
P [kW]	475	1055	475	1055	475	1055
$T_{Coldwater,in}$ [°C]	13	13	13	13	12	12
$T_{Coldwater,out}$ [°C]	8	8	8	8	7	7
$T_{Hotwater,in}$ [°C]	95	95	70	70	95	95
$T_{Hotwater,out}$ [°C]	80	80	60	60	55	55
$T_{Coolingwater,in}$ [°C]	31	31	31	31	31	31
$T_{Coolingwater,out}$ [°C]	36.5	36.5	36	36	36.5	36.5
COP	0.72	0.72	0.4	0.4	0.61	0.61
Op. weight [t]	5.5	10.9	7.7	15.5	7	14.1
Power consumption [kW]	2.4	3.4	3.9	5.8	3.2	4.2
Dimenstions [mm] length/width/height	3680/1244/ 2255	4776/1495/ 2540	3678/2281/ 2084	4776/2795/ 2540	3678/1834/ 2084	4872/2248/ 2519

Table A.3: Technical specifications of chosen machines of York's YIA - series

Parameter	1A1	2B1	5C3	14F3
P [kW]	280	506	1048	3150
$T_{Coldwater,in}$ [°C]	12.2	12.2	12.2	12.2
$T_{Coldwater,out}$ [°C]	6.7	6.7	6.7	6.7
$T_{Hotwater,in}$ [°C]	95	95	95	95
$T_{Hotwater,out}$ [°C]				
$T_{Coolingwater,in}$ [°C]	29.4	29.4	29.4	29.4
$T_{Coolingwater,out}$ [°C]	38.5	38.4	38.5	38.4
COP	0.61	0.69	0.61	0.69
Op. weight [t]	5.182	8.117	13.744	40.837
Power consumption [kW]				
Dimenstions [mm] length/width/height	3740/1680/ 2340	4960/1500/ 2650	6790/1600/ 3030	9310/2400/ 4250

Table A.4: Technical specifications of EAW's Wegracal series

Parameter	**Wegracal SE 15**	**Wegracal SE 50**	**Wegracal SE 150**	**Wegracal SE 200**
P [kW]	15	54	150	200
$T_{Coldwater,in}$ [°C]	17	15	15	15
$T_{Coldwater,out}$ [°C]	11	9	9	9
$T_{Hotwater,in}$ [°C]	90	86	86	86
$T_{Hotwater,out}$ [°C]	80	71	71	71
$T_{Coolingwater,in}$ [°C]	30	27	27	27
$T_{Coolingwater,out}$ [°C]	36	32	32	32
COP	0.71	0.75	0.75	0.75
Op. weight [t]	0.6	2.25	3.6	4.3
Power consumption [kW]				
Dimenstions [mm] length/width/height	1750/760/ 1750	2950/1100/ 2311	3490/1320/ 3000	3490/1320/ 3600

Table A.5: Technical specifications of Shuangliang's single-stage series

Parameter	**99H2**	**165H2**	**265H2**	**331H2**
P [kW]	350	580	930	1160
$T_{Coldwater,in}$ [°C]	15	15	15	15
$T_{Coldwater,out}$ [°C]	10	10	10	10
$T_{Hotwater,in}$ [°C]	95	95	95	95
$T_{Hotwater,out}$ [°C]	85	85	85	85
$T_{Coolingwater,in}$ [°C]	32	32	32	32
$T_{Coolingwater,out}$ [°C]	38	38	38	38
COP	0.81	0.81	0.81	0.81
Op. weight [t]	7.6	9.7	13.9	15.8
Power consumption [kW]	3.8	4.1	6.8	7
Dimenstions [mm] length/width/height	3870/1506/ 2239	3858/1668/ 2541	4420/1784/ 2701	4535/1983/ 2860

Table A.6: Technical specifications of Zephyrus' SAB-HW and SAB-LW machines

Parameter	SAB-HW 003G1	SAB-HW 028G1	SAB-LW 006G1	SAB-LW 028G1
P [kW]	105	985	229	985
$T_{Coldwater,in}$ [°C]	13	13	13	13
$T_{Coldwater,out}$ [°C]	8	8	8	8
$T_{Hotwater,in}$ [°C]	95	95	95	95
$T_{Hotwater,out}$ [°C]	80	80	55	55
$T_{Coolingwater,in}$ [°C]	31	31	31	31
$T_{Coolingwater,out}$ [°C]	36.5	36.5	36.5	36.5
COP	0.71	0.71	-	-
Op. weight [t]	2.4	9.9	7.6	18.1
Power consumption [kW]	3.1	10	4	17.1
Dimenstions [mm] length/width/height	2276/1723/ 1691	5058/1727/ 2530	2897/2084/ 2584	5051/2536/ 2960

Table A.7: Technical specifications of Thermax absorption chillers

Parameter	Cogenie LT-5	Extended Cogenie LT-16C	LT-T (WIN) (Cutechill) 5G 4K C
P [kW]	150	484	960
$T_{Coldwater,in}$ [°C]	12	12	12
$T_{Coldwater,out}$ [°C]	7 (min 4.5)	7 (min 4.5)	7 (min 4.5)
$T_{Hotwater,in}$ [°C]	90 (min.75, max.120)	90 (min.75, max.120)	90 (min.75, max.120)
$T_{Hotwater,out}$ [°C]	80 (min.65)	80 (min.65)	80 (min.65)
$T_{Coolingwater,in}$ [°C]	29 (min.20)	29	29.4
$T_{Coolingwater,out}$ [°C]	34	35.4	36.4
COP	0.68	0.71	0.75
Op. weight [t]	4	7.3	14.9
Power consumption [kW]	2.15	2.55	2.55
Dimenstions [mm] length/width/height	2350/1350/2350	4700/1500/2520	4660/2090/3050

Table A.8: Overview over the data gathered in the inquiry (Heat source: CHP)

100kW			
	Man1	**Man2**	**Man3**
COP	0.71 0.75	0.77 0.84	0.72 0.73
Cold water temperatures	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C
Recooling temperatures	25°C/32°C 25°C/32°C	25°C/32°C 25°C/32°C	25°C/32°C 25°C/32°C
Hot water temperatures	88°C/75°C 87°C/75°C	95°C/75°C 95°C/75°C	95°C/75°C 95°C/75°C
El. power consumption	5.1kW 5.1kW	3.1kW 3.1kW	3.2kW 3.2kW
Spec. investment costs	526€/kW 526€/kW	624€/kW 507€/kW	442€/kW 442€/kW
500kW			
COP	0.75 0.78	0.77 0.84	0.72 0.73
Cold water temperatures	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C
Recooling temperatures	25°C/32°C 25°C/32°C	25°C/32°C 25°C/32°C	25°C/32°C 27°C/34°C
Hot water temperatures	95°C/75°C 95°C/75°C	95°C/75°C 95°C/75°C	95°C/74°C 95°C/73°C
El. power consumption	6.9kW 6.9kW	6.3kW 4.1kW	5.8kW 4kW
Spec. investment costs	213€/kW 191€/kW	224€/kW 186€/kW	199€/kW 191€/kW
1000kW			
COP	0.79 0.82	0.77 0.84	0.72 0.73
Cold water temperatures	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C	12°C/6°C 18°C/12°C
Recooling temperatures	25°C/32°C 25°C/32°C	25°C/32°C 25°C/32°C	25°C/32°C 27°C/34°C
Hot water temperatures	95°C/75°C 95°C/75°C	95°C/75°C 95°C/75°C	95°C/74.6°C 95°C/72.9°C
El. power consumption	6.9kW 6.9kW	11.3kW 10kW	7.3kW 7.3kW
Spec. investment costs	155€/kW 142€/kW	156€/kW 137€/kW	122€/kW 122€/kW

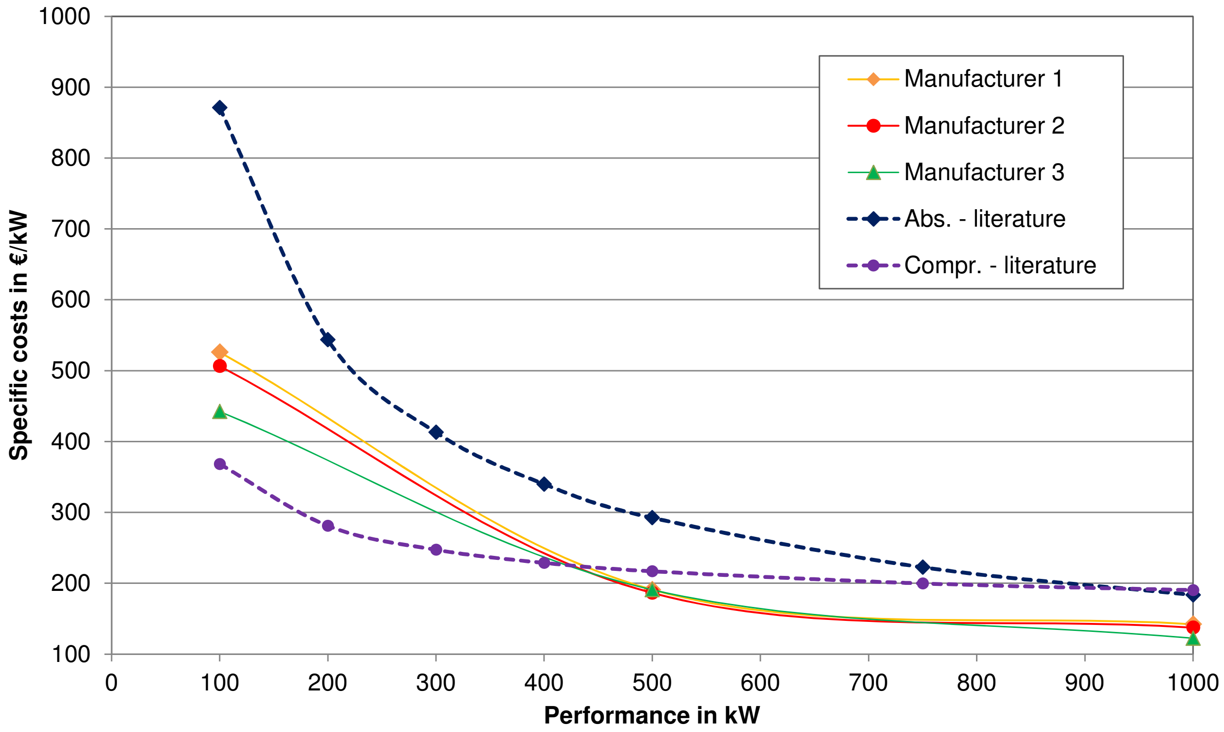

Figure A.1: Specific investment costs of absorption chillers, (Application 1, CHP)

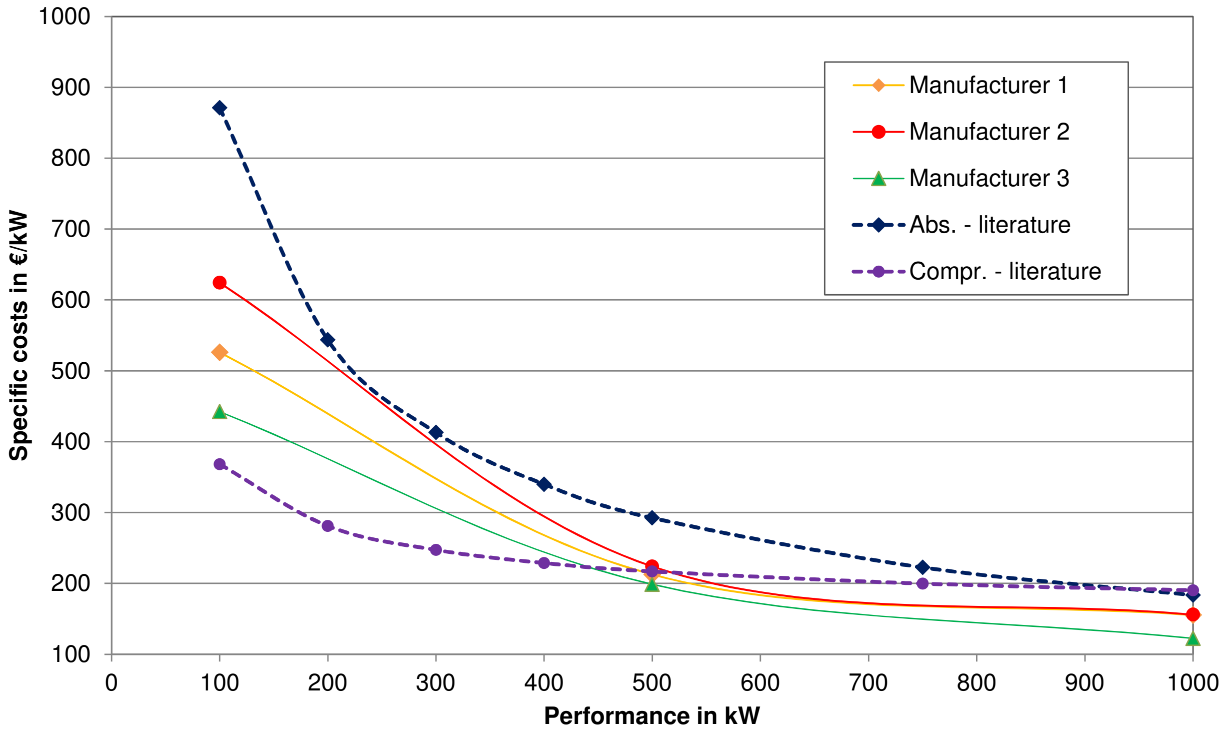

Figure A.2: Specific investment costs of absorption chillers, (Application 2, CHP)

Refrigerant	Composition	Application	ODP (R11=1)	GWP (CO2=1)	Boiling point (°C)	Vapour pressure at 50°C (bar abs.)
HCFCs low chlorine						
R22	$CHCLF_2$	HT, MT, LT	0.05	1500	-41	19.4
R22 Blends	R22 + HFCs	MT	0.03 to 0.05	970 to 1770	-33 to -35	13 to 14
R22 Blends	R22 + HCFs	LT	0.02 to 0.03	1960 to 3570	-44 to -51	20 to 25
HFCs chlorine free						
R134a	CF_3CH_2F	HT, MT	0	1300	-26	13.2
R404A	R143a/125/134a	LT	0	3260	-47	23.0
R407C	R32/125/134a	HT	0	1525	-44	19.8
R410A	R32/125	HT	0	1725	-51	30.5
Other R32 blends	R32 + HFCs	LT	0	1770 - 2280	-46 to -48	21 to 23
Other R125 blends	R125 + HFCs	HT, MT, LT	0	1830 - 3300	-43 to -48	18 to 25
HCs halogen free						
R290	C_3H_8 propane	HT, MT	0	3	-42	17.1
R1270	C_3H_6 propylene	LT	0	3	-48	20.6
R600a	C_4H_{10} isobutane	MT	0	3	-12	6.8
R290 blends	R290 + HCs	HT, MT, LT	0	3	-30 to -48	10 to 18
Other halogen free						
R717	NH_3 ammonia	LT (MT,HT)	0	0	-33	20.3
R744	CO_2 carbon dioxide	HT, MT, LT	0	1	-57*	74**

**Triple point (5.2 bar abs).*
*** At critical temperature 31°C.*

Figure A.3: Properties of refrigerants (values from [4])

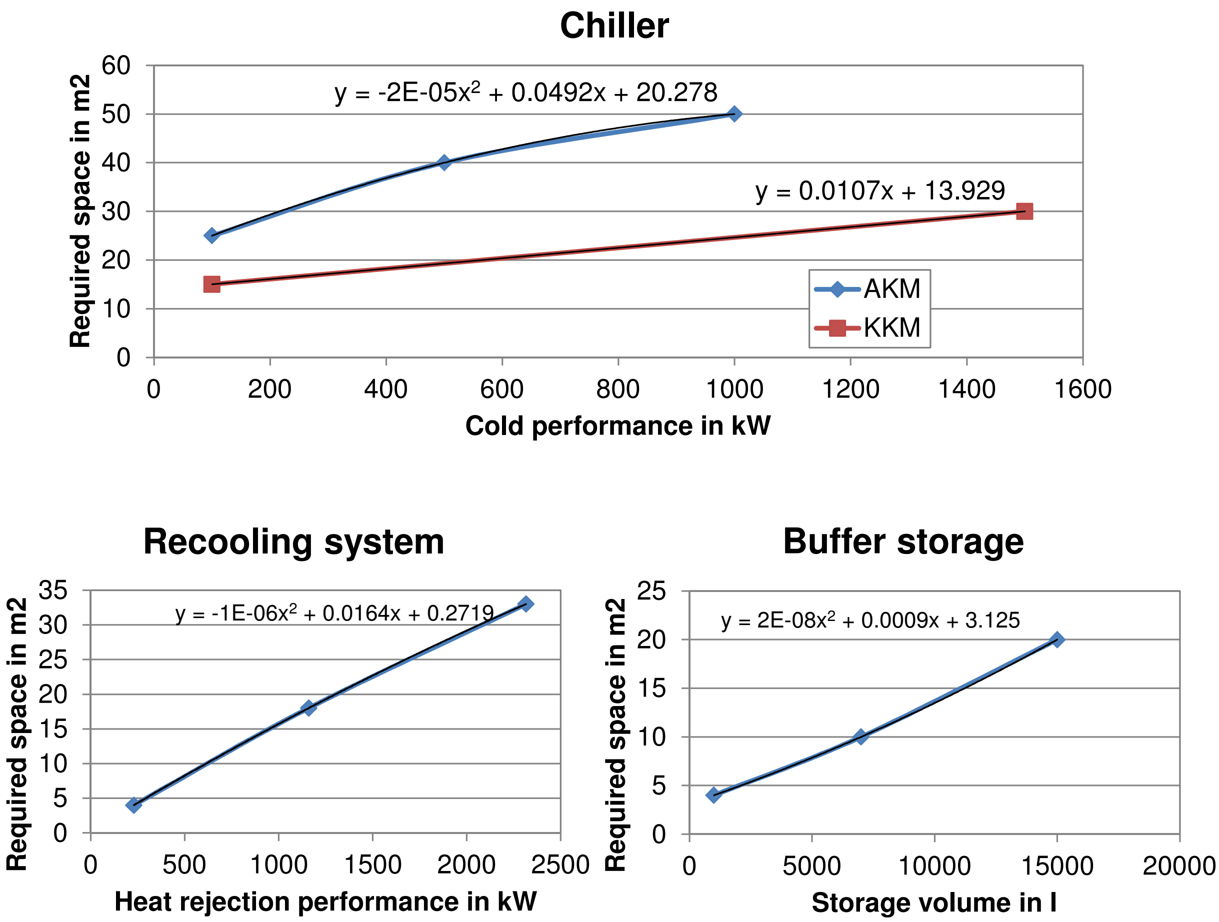

Figure A.4: Estimated space requirements for chillers, recooling systems and buffer storages